CHAPTER 7

EFFECT OF VARIOUS FACTORS ON THE KINEMATIC STRUCTURE OF MACHINE TOOLS

The typical kinematic structure of any machine tool complies in the main with requirements of a kinematic nature, and does not always fully meet other service requirements, especially those concerned with accuracy and production capacity. These requirements are usually met, for the most part, by a suitable construction. In certain cases, however, better results can be attained when definite modifications are made in the typical kinematic structure of the machine tool.

Let us consider these possibilities for changing the typical kinematic structure in accordance with the processing purpose of the machine tool, its universality, the level of the requirements made to the machining accuracy and output, dynamic factors associated with operation of the machine tool, and requirements made to the setting-up facilities.

Depending upon the processing requirements, machine tools of the same processing group may have different kinematic structures. The group of boring machines can be cited as an example. If in boring a workpiece, end flanges must be faced in the same setting, a facing slide is provided on the faceplate. This slide has cross feed and carries a single-point tool called, in this case, a fly cutter. The kinematic chain that powers tool cross feed (Fig. 71*a*) includes a hidden summation drive resulting from gear *c* rolling about gear *b* since the slide is mounted on the rotating faceplate. This summation drive imparts supplementary radial motion to the facing slide and the fly cutter so that the actual radial feed does not comply with the normal expected rate. To compensate for this additional displacement, a differential (Fig. 71*b*) is built into the feed group. Two motions are transmitted through the differential: one is a slow radial feed motion through change gears i_s while the other is a rapid motion, along gear train $1\text{-}4\text{-}5\text{-}6\text{-}\frac{a}{b}$, for compensating for the surplus radial displacement due to the hidden planetary transmission. Thus the kinematic structure of the boring machine has become more complex with the application of a fly cutter.

In addition to the ordinary elements and mechanisms required to bore holes, the design of a jig borer incorporates very complex devices for reading off the co-ordinate dimensions. These devices are required to enable the

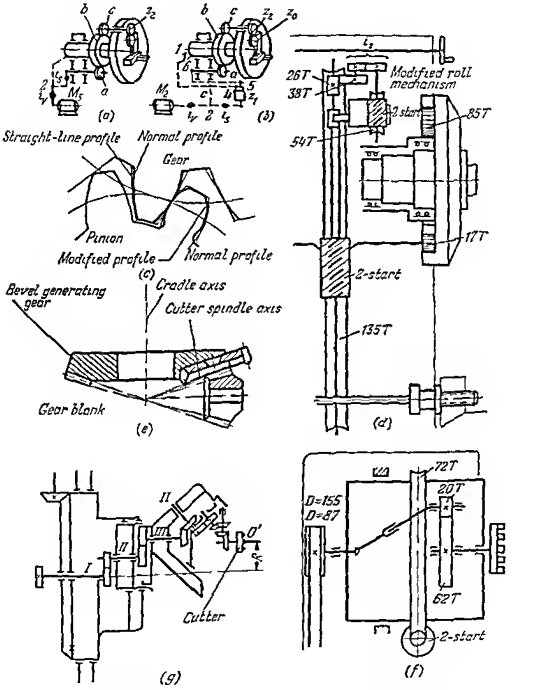

Fig 71 Modifications of the typical structure of various machine tools

holes being bored to be positioned at the specified co-ordinates, and greatly alter and complicate the kinematics of an ordinary boring machine.

Still another example can be mentioned in which the typical kinematic structure is modified to meet the processing requirements.

If a spiral bevel gear generator is to be used to cut Formate gears (Fig. 71*c*), in addition to the conventional spiral bevel gears, then the typical kinematic structure must be changed. The changes that are to be made depend upon which of the two methods for cutting Formate gears is used. When the first method is applied, the cradle has a nonuniform motion for cutting the pinion member (smaller of the pair), such motion being provided by the modified roll mechanism built into the generator (Fig. 71*d*). This nonuniform rotation of the cradle is obtained as a result of two motions of the cradle worm: uniform rotation and nonuniform axial motion transmitted by the modified roll mechanism through its change gears i_z. The second method of cutting a Formate pinion involves the setting of the face-mill type cutter at an angle to the cradle axis (Fig. 71*e*). In this case, the pinions are cut by means of a bevel generating gear and not a crown gear as ordinarily. Motion can be transmitted to the inclined cutter through shafts that are not in alignment, either by means of a telescopic shaft with universal joints (Fig. 71*f*) which are not capable of transmitting heavy torques or, through further complication of the drive, enabling motion to be transmitted by ordinary gearing between nonaligned shafts (Fig. 71*g*). A drive of the second type can easily accommodate heavy loads. In either case, the kinematics of the generator, as far as the drive to the cutter is concerned, differs from the typical structure of a spiral bevel gear generator. Many other examples could be cited showing the further development of the kinematic structure when more extensive demands are made to the processing capacity of a machine tool. This tendency is quite clear however from the preceding examples.

The kinematic structure of a machine tool may be expanded and made more complex in cases when it is necessary either to extend the processing capacities or to increase the output by increasing the number of spindles, clamping stations or sections, or when the machine is to be built into an automated production line or transfer machine.

When the processing capacities of a machine tool are extended, it becomes a multiple-structure machine in which several different combinations of formative, indexing and feed-in operative motions can be obtained. This enables work of various shapes to be machined by various cutting tools in performing different processing operations. By and large, it can be said that the kinematic structure of an omniversal machine tool allows for the performance of all the required operations. If only one operation is being performed, only a part of the machine with its partial structure is being utilized. Hence, provision must be made for devices that enable the machine

to be changed over from one partial structure to another Various procedures are employed in such multiple-structure machine tools to obtain several different partial structures

There are three main procedures for changing over from one partial structure to another The first of these involves the use of the available kinematic operative members one after another without removing any from the machine which, consequently, does not have a single interchangeable unit The second procedure consists in changing, not only various members but whole units of the machine Both of the preceding methods are employed in the third procedure, in which interchangeable units are utilized only partially

An example of a multiple structure machine tool is one for producing globoid worm gearing Model 549, for instance consists of twelve partial structures It can be used to machine globoid worms and their conjugate worm wheels by means of cutter heads, hobs, shaving cutters, grinding wheels and laps, and to perform both roughing and finishing operations This machine has various special devices by means of which any particular partial structure may be set up for operation Such devices include for example a number of jaw clutches In addition to these interstructure devices the machine has three hobbing and grinding attachments which are used in conjunction with one of the available partial structures

Machine tool structures may sometimes differ to some extent if the machine tools are of the multiple spindle or multiple-station type in which several workpieces are machined simultaneously Here new blanks are loaded and the finished work is removed simultaneously with the machining of other blanks In these machines the kinematic structure is made more complex by the use of parallel kinematic chains, so that a multiple-station machine is obtained

Parallel stations of a machine several spindles or slides are sometimes designed in the form of separate sets or units having separate, independent but repeated kinematic structures in which case the machine tool is of the multiple-section type As a rule, all the sections are mounted on a common bed, or base Machine tools of this type can handle identical or different parts simultaneously An example is the two section gear hobbing machine having two sections identical in kinematics and construction mounted on a common base

The general kinematic structure of all multiple spindle multiple-station and multiple-section machine tools is complicated by the provision of supplementary identical kinematic groups and intergroup kinematic constraints

Machine tools based on a diversified structure i e multiple structure types cannot be placed into any definite class of typical kinematic structures Therefore the general structure of such machines is determined by

the class of the partial structure which produces formative operative motions composed of the maximum number of elementary motions.

When a machine tool is built into a transfer machine or automated production line, the kinematic groups for the controlling motions are considerably expanded and become much more complex. As a result, the kinematic structure of each separate machine tool may also be changed.

It is hardly expedient to take up the kinematic structure of automatic transfer machines in the present part of this textbook. Nevertheless, the theoretical principles formulated above are applicable in analysing the kinematic structure of a transfer machine if it is considered as a whole, i.e. as a separate unit. The kinematic structure of any component machine tool, then, consists of intragroup and intergroup kinematic constraints, which are to be regarded as intramachine constraints in relation to the whole transfer machine. Consequently, a transfer machine, or automated production line, can be considered to be made up of intramachine and intermachine constraints. The latter possess many specific features, and will be taken up in Part Six of Vol. 4.

Higher machining-accuracy requirements may be met to some extent by changing the typical kinematic structure along the following lines: improving the structure of the internal kinematic constraints; increasing the number of elementary motions making up complex operative motions, correction devices being used for this purpose; and by applying nonmechanical internal constraints. The accuracy of internal kinematic chains can be increased by using spur gears in them instead of helical or bevel gears, or other types of gearing on intersecting or crossed axes. This principle can be demonstrated by the example of the gear-shaping machines, models 5A12 and 5B12.

These two models are very similar in their capacities, quantity and type of kinematic groups, arrangement and construction features. They differ in the number and type of members that compose the internal kinematic indexing chain with change gears i_x (Fig. 72), interconnecting shaping cutter rotation with blank rotation. Both models have precision worm gearing units for driving the cutter ram and work spindle, but the train between the worm shafts of these drives includes four pairs of bevel gears and two other pairs of gears in model 5A12 (Fig. 72), while eleven pairs of spur gears are provided in this train of model 5B12 (Fig. 73). The second model has no bevel gearing.

The substituting of spur gears in the indexing train of model 5B12 for bevel gearing enables this gear shaper to produce gears of a higher accuracy class than those made by model 5A12.

However, this change in the structure of the indexing gear train has led to an increase in the number of gears (model 5A12 has 16 gears while model 5B12 has 21), but the accuracy of the indexing train has been substantially raised in model 5B12. The kinematic accuracy of the latter model

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PART THREE

KINEMATICS OF MACHINE TOOLS

A VAST NUMBER OF METAL CUTTING MACHINE TOOLS (literally millions) are being employed in the world engineering industries. The rapid advance of science and engineering has led to the demand for a further extension of the type-and-size range of machine tools. The wide automation of machine tool controls has created prerequisites for the versatile application of electric, hydraulic, pneumatic and other devices in machine tool design, thus complicating the kinematics and construction of these up to date models.

The design of any machine tool is based primarily on its *kinematic structure*, expressed in terms of its kinematic scheme. For the sake of brevity this can be called the *kinematics* of the machine tool. Consequently a thorough understanding of the kinematic structure is a first essential both in the design and operation of machine tools. Notwithstanding the great diversity of machine tools intended for performing not only different but even similar operations, the kinematic structure of any machine tool is based on a number of fundamental, sufficiently general kinematic principles that all machine tools comply with. A mastery of these principles contributes to the development of new metal cutting machine tools that best fill the requirements made to their accuracy and production capacity. It also enables the lead time of a new model to be substantially reduced and the finished machine tools to be more rationally employed.

Because of their great variety, it is frequently very difficult to gain an understanding of the construction of a machine tool and therefore of its mode of operation and processing capacities without sufficient knowledge of the general procedures used in kinematic analysis.

Part Three treats of the interrelations between the workpiece to be produced in the given machine tool, the cutting tool to be used, the selected processing method and the kinematic structure of the machine tool. Typical kinematic schemes are presented and a method is given for analysing the kinematic structure of machine tools whose design may be based upon any kind of kinematic constraints. This general procedure enables a kinematic scheme of any complexity whatsoever to be reduced to one of

the comparatively few standard schemes whose investigation offers no difficulties.

All the theoretical propositions of machine tool kinematics follow from the kinematics involved in shaping the surfaces being machined. Therefore, kinematic analysis begins with an investigation of the surfaces obtained in machining as performed by metal-cutting machine tools of various types and the methods of shaping them. For the purposes of this book, shaping is considered to mean the imparting of a definite shape to a surface, and not the operation performed by the machine tool called the shaper.

CHAPTER 1

SHAPING SURFACES

1-1. Surfaces Obtained by Machining

The *geometrical surfaces of* machine parts and tools are extremely diverse Surfaces of practically any required shape can be produced in machine tools, employing various cutting processes and suitable cutting tools The most readily machined surfaces are planes, circular cylinders and cones, helicoids, as well as cylindrical and conical surfaces shaped on the basis of the involute of a circle, certain spirals, etc These are all surfaces that can be produced by a *combination of rotary and rectilinear uniform motions of the cutting tool* and work It is these motions that are most easily realized by means of simple mechanisms

The following classification of surfaces is found to be convenient in investigating machine tool kinematics and in solving problems (1) planes, (2) circular and noncircular cylinders, (3) circular and noncircular cones (4) single-curved and warped ruled surfaces, (5) spherical surfaces, and (6) double-curved surfaces.

If these surfaces are regarded as the trace produced in the motion of one generating line—the generatrix—along another line—the directrix—then [illegible]

(2) those produced by one variable and one invariable line, and (3) those produced by two variable lines

If the functions of the generatrix and the directrix can be interchanged in producing a surface i e the generatrix can be used as the directrix and vice versa, without changing the shape of the surface (for example, a cylindrical surface), such a surface is said to have invertible generating lines An example of a surface with noninvertible generating lines is the circular cone If the base circle of the cone is taken as the generatrix and is moved along a straight line (directrix) no cone is produced

Surfaces may be enveloping, or internal, or they may be enveloped or external However, not all surfaces can be differentiated by these designations For example the side surfaces of gear teeth can be classed as enveloped (external) surfaces However, if the tooth space of a gear is regarded as a single continuous surface, these same surfaces must be classed as enveloping (internal) surfaces

Surfaces can also be classified as closed (for instance, a circular cone, elliptic cylinder, complete sphere, etc.) or open (an incomplete spherical surface, plane, helicoid, etc.).

Workpieces that are to be machined rarely have only one surface as, for example, the balls of ball bearings. In the great majority of cases, workpieces are bounded by several surfaces which must be in strictly definite positions in respect to each other. In such an event, the workpiece has edges which are the conjunction lines of the surfaces. In machining a workpiece it is necessary to obtain the required surfaces in their proper relative positions. The whole surface of the workpiece is made up, therefore, of a series of elementary surfaces. For instance, a spur gear can be conceived as two sets of identical cylindrical surfaces (with an involute directrix), definite in number, arranged symmetrically about the gear axis and constituting the right and left sides of the teeth. The top land of each tooth is an open surface of a circular cylinder while the surface at the bottom of the tooth spaces is also an open cylindrical surface with a directrix of some specified form. A multiple-start cylindrical worm is a complete set of cylindrical and helical surfaces located symmetrically about a single axis—that of the worm.

Sometimes a surface, especially if it is of great extent, is conditionally broken down into a series of elementary surfaces, each of which is machined separately in the machine tool.

The classifications of surfaces given above enable any surface to be appraized in respect to the possibility of shaping it in a machine tool, without determining the actual parameters of the geometrical shape concerned. Thus, it is easier to machine a surface with invariable generating lines than one with variable lines. A greater number of methods are available for machining closed surfaces of the same type (for instance, a complete circular cylinder) than for open surfaces (for example, an incomplete circular cylinder cannot always be machined in a lathe). The same is true of invertible surfaces; they can be produced by a greater number of types of cutting tools (in respect to the shape of the cutting edge) than surfaces with noninvertible generating lines.

The geometrical shape of most engineering surfaces, i.e. geometrical surfaces used in the various branches of engineering, can be shaped by using generating lines of the following types.

A. Lines realized in machine tools by means of simple—rotary and rectilinear—and only uniform motions, such as: (1) straight lines, (2) circles or their arcs, (3) involutes of circles (normal, curtate and prolate types), (4) helixes on cylinders, cones and globoids, (5) spirals of Archimedes, (6) epicycloids (normal, curtate and prolate types), (7) hypocycloids (normal, curtate and prolate types), and (8) space curves produced by uniform rotary and rectilinear motions (for example, the curve of the relief surface on hob teeth).

B Lines realized in machine tools by means of both uniform and nonuniform simple—rotary and rectilinear—motions such as (1) parabolas (2) hyperbolas (3) ellipses (4) sine curves and (5) logarithmic spirals

It is by means of these lines that the shape of a surface is specified For example the side surface of the teeth of a helical gear is specified by the form of its generating lines It will be an involute cylindrical helicoid since one generating line is an involute of a circle and the other is a helix on a cylinder If the work piece is bounded by several surfaces of the same type then in addition to specifying the geometrical shape of these surfaces it is necessary to indicate their number and relative positions

In certain cases in particular for surfaces with variable generating lines the form is specified either graphically (by means of drawings or templates) or by an equation or equations employed to construct or plot the cam or template curves

Listed below are the possible and most frequently encountered shapes of parts used in the engineering industries

(1) external and internal bevel gears of the ordinary or hypoid types with involute or noninvolute tooth profiles with uniform depth or converging teeth with straight circle arc or curvilinear teeth

(2) external and internal circular spur helical and herringbone gears and segmental gears with normal full crowned or localized bearing teeth in the form of single gears or double and multiple cluster gears

(3) elliptical spur and helical gears and segmental gears

(4) standard and globoid worm wheels

(5) straight and globoid worms single and multiple start types with the following tooth forms worm cut with straight sided milling cutter with straight flanks in the axial section and involute helicoidal

(6) straight flat racks spur and worm types crown gears with straight or curved teeth complete or segmental and circular racks

(7) contoured surfaces of revolution

(8) straight and taper thread with standard or coarse pitch single or multiple start external or internal with uniform or nonuniform pitch

(9) relief surfaces of cutting tools

(10) internal and external straight sided and involute straight and helical splines

(11) complex contoured surfaces

(12) plate cylinder and single-edge cams

(13) external and internal incomplete and complete spherical surfaces

(14) circular and noncircular complete and incomplete external and internal plain and stepped cylindrical surfaces

(15) circular and noncircular complete and incomplete external and internal tapered surfaces

(16) polyhedral bodies

(17) planes;
(18) wedges and taper gibs;
(19) flat- and V-belt pulleys;
(20) housing-type parts (beds, bases, stanchions, gearbox housings, etc.).

1-2. Shaping Geometrical and Real Surfaces

Real surfaces, obtained on a solid body by any method of working the material (casting, pressworking, machining, metallizing, etc.), have one feature in common, consisting in the fact that any real surface is only a certain approximation of the corresponding geometrical (imaginary or "ideal") surface. Therefore, the processing operation of producing a real surface already incorporates the process of shaping the corresponding geometrical surface or, in other words, incorporates the process of geometrically shaping the real surface.

A *geometrical surface* is usually defined as the trace obtained in the motion of one geometrical generating line, called the *generatrix*, along another geometrical generating line, called the *directrix*. The *trace* is understood to be the shaped surface conceived as a continuum of consecutive geometrical positions of the generating line.

Consequently, to shape any geometrical surface, two geometrical generating lines and their relative motion are required (Fig. 1*a* through *f*).

Real surfaces can be shaped on metal or other materials with the aid of auxiliary bodies having auxiliary real surfaces, lines and points which shall be conditionally referred to, from here on, as *auxiliary material elements* in contrast to the imaginary elements, nonexisting in reality, which are to be called, also conditionally, *geometrical elements*. The geometrical generating lines and, consequently, the required surfaces are produced in the motion of these real auxiliary elements.

The relative motions of the geometrical lines in producing surfaces are called *formative motions.*

Therefore, the shaping of a surface primarily involves the production of the geometrical generating lines, as a result of whose relative formative motions the surface is produced.

There are four methods of producing geometrical lines with an entirely definite number of formative motions.

The *forming method* (Fig. 1*g*) is one in which the configuration and extent, or length, of the auxiliary material line coincide with the configuration and extent of the line being produced. The latter is obtained as a copy or "mirror reflection" of the material line. The geometrical line is produced without any formative motions; all that is required is a positioning motion to move the auxiliary element to the initial position.

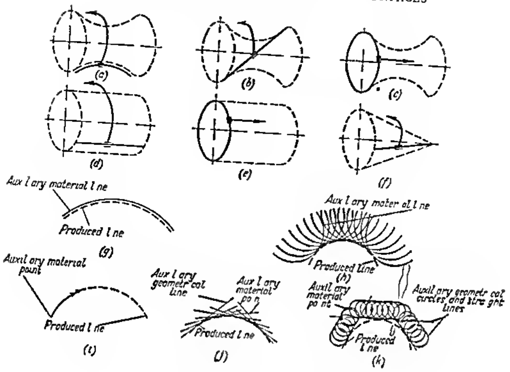


Fig 1 Methods of producing geometrical surfaces

The *generating method* (with one formative motion—rolling) is one in which the line (Fig 1*h*) is obtained as the envelope of the consecutive positions occupied by the auxiliary element (in the form of a line) as it rolls along the line being produced

The *tracing method* (also with a single formative motion) is one in which the auxiliary element in the form of a material point (Fig 1*i*) produces the line being formed as the trace it leaves in its motion (the material point is understood, of course, to be a short length of the cutting edge on the tool)

The *tangent method* (with two or more formative motions) is one in which the line being formed is tangent to a series of supplementary auxiliary lines, produced by the material point either by the tracing method (Fig 1*j*) or the tangent method (Fig 1*k*)

To produce a specified surface, it is necessary to have a geometrical generatrix and directrix of corresponding configuration which can be formed by any one of the four methods outlined above It follows that the methods of shaping surfaces are composed of the methods used to produce the geometrical generating lines of the surface being shaped Since many combinations

of line formation methods are possible, the method of shaping a surface depends, not only on the configuration of the auxiliary element (cutting tool) and the method of producing each separate generating line, but also upon the combination of methods used for the geometrical formation of the generating lines.

Figure 2 illustrates the geometrical methods of shaping various surfaces depending on the combination of methods used in the formation of the geometrical generatrix and directrix. In embossing or forming sheet metal (Fig. 2*a*), both geometrical generating lines are produced by the forming method and therefore no formative motions are needed. One motion for setting the material auxiliary surface into its final position is sufficient.

In rolling surfaces (Fig. 2*b*), in particular in burnishing bed ways with a roll, the generatrix is produced by the forming method (formation of the line along the roll length) while the directrix is obtained by the generating method and therefore one formative motion is required. Here F_1 is the rolling motion of the burnishing roll.

In cutting thread with a form tool (Fig. 2*c*), the configuration of the generatrix is a copy of the formed cutting edge of the tool, while the helical directrix is obtained as the trace in a single formative motion F_v.

When thread is milled with a single-thread cutter (Fig. 2*d*), the generatrix is produced by the forming method and the directrix by the tangent method, the latter requiring two formative motions: rotation F_v and helical motion F_{s1}.

The geometrical generatrix is produced in the four preceding cases by the forming method which requires no formative motions, while the directrix is produced by the forming, generating, tracing or tangent method. Therefore the number of formative motions required to produce the surface is determined only by the method of producing the directrix.

The generatrix and directrix of the surface being shaped are produced in the remaining examples (Fig. 2*f* through *p*) by one or several formative motions and therefore the number of formative motions required to produce a surface will be the sum of the numbers of formative motions required to produce each of the two geometrical generating lines of the surface to be machined. The remaining illustrations of Fig. 2 show the following machining methods and the corresponding methods of geometrically shaping the surfaces.

In Fig. 2*f*, a ball is burnishing a contoured surface of revolution by the double rolling method with two formative motions: F_1 being the rolling of the ball around the circumference and F_2 being the rolling of the ball along the curvilinear contour (profile) of the surface of revolution.

Figure 2*g* illustrates the generation of the side surfaces of the teeth on a spur gear by means of a gear-shaper cutter, employing a combination of the generating and tracing methods with two formative motions: F_1

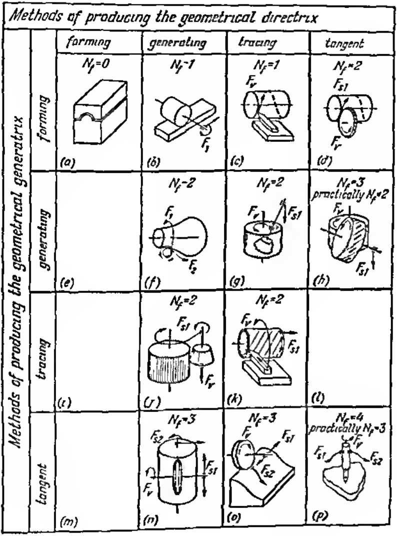


Fig. 2. Possible methods of shaping surfaces

(generating motion made up of co-ordinated rotation of the gear blank and cutter) and F_{s1} (rectilinear motion, or feed, of the cutter).

In Fig. 2*h*, a spur gear is being ground with a helically profiled (worm-type) grinding wheel by a combination of the generating and tangent methods. Three formative motions are required: one motion is generation for shaping the tooth profile, and two motions—grinding wheel rotation and its longitudinal motion along the teeth—are required to shape the tooth form along its length. Since it is necessary to rotate the wheel to obtain the generating motion of the standard rack, this rotation is used to shape the tooth form along its length. Consequently, two of the three formative motions coincide and the gear is ground with two motions F_v and F_{s1}.

In Fig. 2*j* a rotary gear-shaping cutter cuts a spur gear by the tracing and generating method with two formative motions.

The relief surfaces on the helical teeth of a plain milling cutter (Fig. 2*k*) are relieved by a pointed tool by the double tracing method with two formative motions: F_v along an Archimedean spiral and F_{s1} along a helix.

In Fig. 2*n*, a disk-type grinding wheel is generation grinding a spur gear with three formative motions, F_v, F_{s1} and F_{s2}, by a combination of the tangent and generating methods.

A curvilinear surface is ground in Fig. 2*o* with a disk-type wheel by the tangent and tracing method with three formative motions.

The milling of a curvilinear three-dimensional surface with an end mill in a tracer-controlled duplicating machine (Fig. 2*p*) is done by the double tangent method. Each of the curvilinear generating lines is produced by two motions: F_v, and F_{s1} or F_{s2}. To realize this method, four formative motions are required but, since the cutting motion F_v participates in the production of both lines, actually only three different motions are necessary.

Three combinations of line formation methods (Fig. 2*e*, *i* and *m*), generating and forming, tracing and forming, and tangent and forming, are purely theoretical cases that are unrealizable in practice.

The shaping of surfaces by the tracing and tangent method (Fig. 2*l*) has found no application as yet in its "true form".

In the general case, the number of formative motions is determined by the equation

$$N_f = N_g + N_d - \frac{1}{2} N_c \qquad (1)$$

where N_g = number of formative motions required to produce the generatrix of the geometrical surface

N_d = number of formative motions required to produce the geometrical directrix

N_c = number of coinciding formative motions.

On the basis of the methods used to produce geometrical lines, it is possible to theoretically establish the *maximum possible number of formative motions*, which is equal to six. Due to coinciding motions there are no more than three motions in any real case. Thus, a surface may be shaped by machining by one, two or a maximum of three formative motions.

The number of possible methods for shaping surfaces greatly increases if one takes into account that certain surfaces can be shaped by generating lines of various forms and with various relative motions.

If the geometrical shaping of surfaces is considered only in reference to machining, there are considerably less methods (namely, seven, see Table 1).

TABLE 1

Configuration of the auxiliary element (cutting edge)		Methods of geometrically shaping surfaces by machining	Number of formative motions
Material line: coincides with generatrix		Forming and tracing Forming and tangent	1 2
Material line: *does not coincide with* generatrix		*Generating and tracing* Generating and tangent	2 3
Material point		Tracing and tracing (double tracing) Tangent and tracing Tangent and tangent (double tangent)	2 3 4 and more

Therefore, each of the different methods of processing solid materials, such as casting, pressworking, burnishing, moulding, machining, electrical-discharge machining, etc., has its own entirely definite method with which surfaces are geometrically shaped. This interrelation between the processing method and the method of geometrically shaping surfaces is shown in Table 2.

TABLE 2

Interrelation Between the Processing Method and the Method of Geometrically Shaping a Surface

Processing method	Formative tool	Type of auxiliary element	Method of obtaining geometrical generating lines: generatrix	Method of obtaining geometrical generating lines: directrix	Number of formative motions
Casting	Foundry moulds	Surface	Forming	Forming	0
Die forging	Dies	Surface	Forming	Forming	0
Blanking	Dies	Line	Forming	Tracing	1
Knurling and rolling	Knurls or rolls	Line	Forming	Generating	1
Ball burnishing	Ball	Surface	Generating	Generating	2
Metal cutting (machining)	Cutting edge	Line or point	Tangent	Generating, tracing or tangent	1 to 3
Grinding	Grinding wheel	Line or point	Tangent	Generating, tracing or tangent	2 or 3
Honing	Abrasive sticks	Point	Tracing	Tracing	2
Abrasive vapour blasting	Abrasive slurry	Surface	Forming	Tracing	1
Buffing	Buffing wheel	Point	Tangent	Tangent	3
Lapping	Lap	Point	Tracing	Tracing	2
Superfinishing	Abrasive stones	Point	Tangent	Tangent or tracing	3
Ball hole-burnishing	Ball	Line	Forming	Tracing	1
Electrical-discharge machining	Electrode	Line	Forming	Tracing	1
Plastics moulding	Moulds	Surface	Forming	Forming	0
Metallizing	Workpiece	Surface	Forming	Forming	0
Chromium and nickel electroplating to size	Workpiece	Surface	Forming	Forming	0
Ultrasonic machining	Sonic contour	Line	Forming	Tracing	1
Wire-drawing	Drawing die	Line	Forming	Tracing	1

CHAPTER 2

CLASSIFICATION OF MOTIONS

The shaping of surfaces by machining requires a more or less considerable number of kinematic links including, besides those which carry the tools and are actually engaged in metal cutting (and which shall be referred to from now on as *operative members*, or *links*), numerous other links with various functions in the machine tool All of these links have quite definite displacements, in full accordance with the given process

Each of these displacements adheres to laws that *can be specified by special parameters—motion parameters*—which enable the given motion to be appraised and, if several motions are concerned, to compare them to each other

It is evident that all motions must be defined, or specified, by space and time parameters

Space parameters include (Fig 3) path, path length, velocity, direction, and the initial point—the beginning of motion

In addition to these motion parameters, characterizing the motion itself, it is also necessary to determine the geometrical position of the given path in space, relative to the paths of other moving bodies and to a stationary point of reference

Each *motion parameter* has its *qualitative and quantitative aspects* The quantitative aspect shall be understood as the initial value or position of the parameter, the qualitative aspect as the mode of variation of this initial value, i e *the law governing the variation of the quantitative aspect* of the parameter In a particular case there may be no variation, i e the quantitative aspect of the motion may remain constant

Motions are assessed by two time parameters

(a) initial moment of the motion either absolute or relative which characterizes the position of the given motion in the cycle or sequence of motions,

(b) the nature of the motion in the course of time in the sense of its continuity motion within the limits of a given path length may be either intermittent, with various laws of intermittence or continuous

Thus, if a motion is considered separately not in relation to other motions, it must be specified by a great number of space and time parameters In the case of a system (set) of moving bodies, the parameters will be even more numerous since the motions must be co ordinated both in space and time

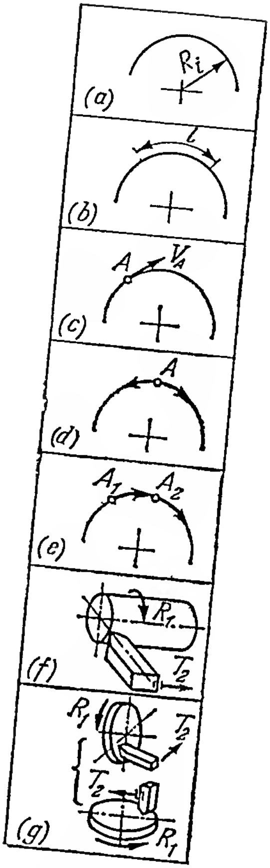


Fig. 3. Motion parameters:
(*a*) path; (*b*) path length; (*c*) velocity; (*d*) direction of the motion; (*e*) initial point on the path; (*f*) relative geometrical position of the path; (*g*) absolute geometrical position of the path

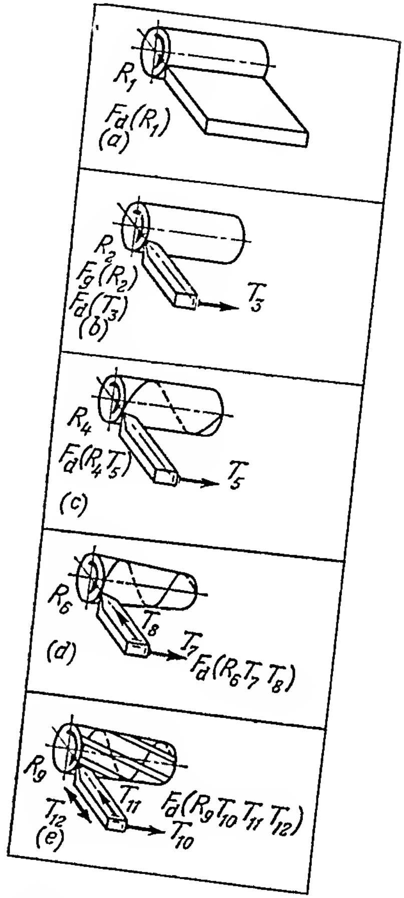


Fig. 4. Simple and complex operative motions in shaping surfaces:
(*a*) and (*b*) turning a cylinder with a wide tool and with a pointed tool, respectively; (*c*) and (*d*) cutting straight and taper thread, respectively, with a single-point tool; (*e*) relieving a taper tap with a single-point tool

Motions are characterized, not only by their space and time parameters, but by their specific functions in the system of motions. In machine tools, as in other industrial machinery, each motion of the machine serves to accomplish some definite production function. Such motions are called *operative motions*. The path of such motions may be very simple—a circular arc or a straight line—but may be of more complex configuration, namely, a helix, involute of a circle, space spiral, etc. In the latter cases, a single rotary or a single rectilinear motion proves insufficient to move the machine component along the given complex curve. Several rotary and/or rectilinear

motions may be required to produce an operative motion along the specified path. Neither of these rotary or rectilinear motions, taken separately, can provide a solution for the production problem for which the operative motion is *intended*. Therefore, motions which make up operative motions are called *elementary motions*. Their path is most frequently obtained by means of rotary or rectilinear translational pairing elements.

These elementary motions compose the complex operative motions. Figure 4 illustrates examples in which *surfaces are produced by a single* complex operative motion composed of several elementary motions. Thus straight thread is cut with a single-point tool in one two element motion $F_d\ (R_4T_5)$ as shown in Fig. *4c*, while taper thread is cut in a single three-element motion $F_d\ (R_6T_7T_8)$ as in Fig. *4d*.

In principle, any number of elementary motions can compose one operative motion. Operative motions are encountered in practice that comprise four or five elementary motions. For example, in relieving taper tap threads with a single point tool, the formative motion consists of four interrelated elementary motions (Fig. *4e*). Such complex motions, however, are comparatively rare.

Only work rotation is required to turn a cylinder with a wide tool (Fig. *4a*) and therefore the operative motion $F_d\ (R_1)$ is of the simple, single element type. The same cylinder is turned with a pointed tool (Fig. *4b*) by two operative motions $F_g\ (R_2)$ and $F_d\ (T_3)$, each of which is also a simple single-element motion.

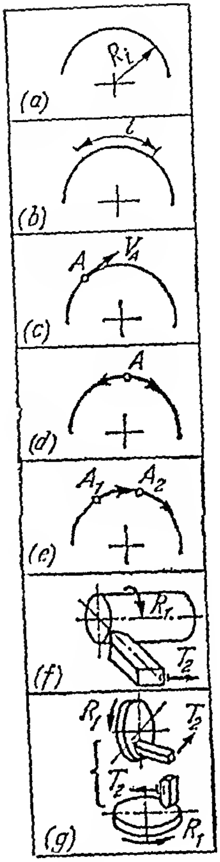
Two features are characteristic of elementary motions that make up definite operative motions.

(1) They are always simultaneous, i.e. they begin at the same moment of time and are of identical duration. This circumstance enables the path length and velocity of an operative motion, composed of several elementary motions, to be appraised, for all practical purposes, on the basis of the path length and velocity of one elementary motion.

(2) The parameters of elementary motions composing an operative motion are always interconnected by definite relationships.

If an operative motion is of the simple type, i.e. consists of a single elementary motion, then the parameters of the latter will become the parameters of the operative motion (Fig. *4a* and *b*).

If a motion is complex, consisting of several elementary motions, its parameters depend upon the parameters of all the elementary motions of which it is composed. Any variations in the ratios of the parameters of elementary motions change the path of the operative motion. Changes made in the absolute values of the parameters of the elementary motions, their ratios being maintained constant, lead to changes in the absolute value of the corresponding parameter of the operative motion with the exception of the parameters of the path.



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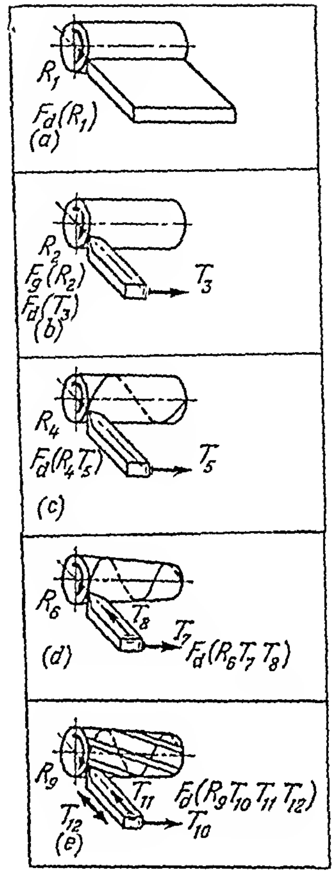

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Operative motions are differentiated by the degree of variability of the initial values of the parameters in the course of motion. Thus, they can be classified as operative motions with either constant or variable parameters.

In addition to the preceding classification, operative motions can also be divided in accordance with the controllability of the quantitative and qualitative indices of the motion parameters. Here, controllability of the parameters is understood to mean the possibility of setting up and obtaining parameters of various initial values on the machine tool (quantitative controllability) which can be varied according to various laws in the course of motion (controlling the qualitative index of the parameter). Operative motions may have all controllable, all uncontrollable or certain controllable and certain uncontrollable parameters.

Operative motions must also be differentiated as to their functional purpose. These motions can be primarily divided into two groups: motions which move the cutting edge of the tool in reference to the work, and motions which do not move the cutting edge and, therefore, do not directly participate in the machining process, but are of a supporting nature.

Motions of the first group can be further divided into three subgroups: *formative*, or *shaping*, *motions*, *indexing motions* and *positioning motions*.

If cutting occurs during a positioning motion, it is called a *feed-in motion.*

All positioning motions in which no cutting occurs are called *setting-up motions.*

Motions of the second subgroup, supporting the motions of the first subgroup, are divided into *controlling motions* and *handling motions.*

Operative motions can also be of the continuous or intermittent (periodic) type.

In machine tools, the formative, or shaping, process is accomplished by metal cutting. There is, therefore, an intimate relationship between the concepts "formative motion" and "cutting motion".

The cutting speed and feed motions are formative motions, and conversely any formative motion is either a cutting speed or cutting feed motion.

In accordance with the above-indicated features, the characteristics of any operative motion can be denoted conventially by a combination of letters as, for example,

$$F_v(R_1T_2T_3) \quad n_1 = \frac{L_2}{P} \text{ and } L_3 = L_2 \tan\alpha$$

which is used in cutting taper thread with a single-point tool.

The letter F indicates that the operative motion is a formative one, and the subindex v indicates that, in reference to the cutting process, it is at the same time a cutting speed motion. If, on the other hand, it is a cutting feed motion, the subindex will be s. The parentheses following the letter F show what elementary motions comprise the operative motion. If the ele-

mentary motions are uniform in nature, the ratio of their velocities or path lengths follows the parentheses

In the preceding example n = rotational speed in rpm; L = displacement in rectilinear motion, P = pitch (or lead) of the thread being cut, and α = angle of taper of the thread (one half of the included angle). If one of the elementary motions is nonuniform, the nonuniformity sign is written above the letter representing this motion as, for example $F_v\ (R_1T_2\overset{\neq}{T}_3)$ and, if possible, the law of the relative nonuniformity is indicated following the closing parenthesis

The following notation has been accepted for nonformative operative motions, if they are simple operative motions, such as the indexing motion *Ind* (R_4), feed-in motion *FI* (T_5), controlling motion *Ct* (R_6) and handling motion *Hd* (T_7).

CHAPTER 3
KINEMATIC CONSTRAINTS

A kinematic interlink, or constraint, is required between the operative members (links) of a machine tool, and between the operative members and the source of motion to produce an entirely definite operative motion in the machine tool. These two kinematic constraints comprise the *kinematic group.*

The kinematic group can be defined as the mechanism (device) which produces a single operative motion with given parameters determining the path, path length, velocity and direction of the motion, as well as the position of the beginning of the motion and the relative geometrical position of the path of the motion being produced.

The structure of any kinematic group always consists of internal and external kinematic constraints. An internal constraint, providing for the given path of the operative motion, may consist of only a single kinematic pair, as in a *simple group*, or of several pairs and kinematic chains linking the movable members of these pairs together, as in a *complex group*. In the latter case, the number of internal kinematic chains which make up an internal kinematic constraint is one less than the number of elementary motions composing the operative motion produced by the group.

Of course, no internal kinematic constraint can produce an operative motion by itself, without a source of motion. Its purpose is to co-ordinate the parameters of the elementary motions in such a way that the operative motion would proceed along the given path if motion occurs in the internal constraint.

Thus, an *internal*, or *operative*, *constraint* is a kinematic constraint which provides the required path of the operative motion. A kinematic group, however, also requires motion produced by some source, which separately powers the given group or is common for the whole machine tool. The most widely used sources of motion in metal-cutting machine tools are electric and hydraulic motors; air and other motors are less commonly employed. Without considering the construction and principle of these motors, this not being an objective of machine tool structural analysis, it is necessary to point out only the fact that each motor consists of two parts: one to which power is supplied and the other in which the motion of the output member is originated. From here on, the first part of the motor will be differentiated and conditionally called its *power part* and the second, the *mechanical part*

of the motor The initial motion is taken off the output member of the motor and is transmitted further along the external constraint (along the kinematic chains of the drive) to the internal constraint of the group

A kinematic or operative group of such structure, producing a complex two element operative shaping motion F_1 (R_1R_2) is illustrated in Fig 5a The external (section *1-3*) and internal (section *1-2*) kinematic constraints are denoted by single dash lines which indicate that the kinematic constraint is accomplished through mechanical members constituting the kinematic chain The diamond symbols denote setting up units

Double dash lines can also be found in Fig 5 They indicate that the kinematic constraints between the operative members and the source of motion are accomplished, not through mechanical members, but through other elements transmitting power pulses which are transformed into mechanical power again in the motors

To this end, kinematic groups incorporate devices called *transducers* or *generators*, in which mechanical motion is transformed into power pulses In diagrams, such devices are conventially denoted by triangles with single or double dash lines, as shown in Fig 5b where a mechanical constraint is connected to transducer Tr_1 at point *3* and an electrical constraint at point *4*

An electric or hydraulic tracer-controlled slide on a lathe can serve as an example of a transducer in the sense here employed In these slides, the motion of the tracing stylus produces or changes an electrical pulse which is reconverted into mechanical motion again in the appropriate devices In the diagram of Fig 5b, for example, a power signal from the transducer is transmitted to the motion metering device which it controls A motion metering device is one which allows the mechanical motion delivered to it to be transmitted further, not completely but only in part Only a definite "amount" or "portion" of motion is discharged, the rest, unused motion is lost in the device Transducers and metering devices, like motors, have power and mechanical parts

In Fig 5, metering devices are denoted by circles with two mechanical constraints (single dash lines) and one power constraint (double dash lines)

Kinematic groups with mechanical constraints (Fig 5a) are extensively employed in machine tools Examples of the structures of such groups with a single initial motion (single source of motion) are to be considered in detail in Part Three

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of the operative motion are due to internal and which to external constraints The internal mechanical constraint provides the path of the motion, the

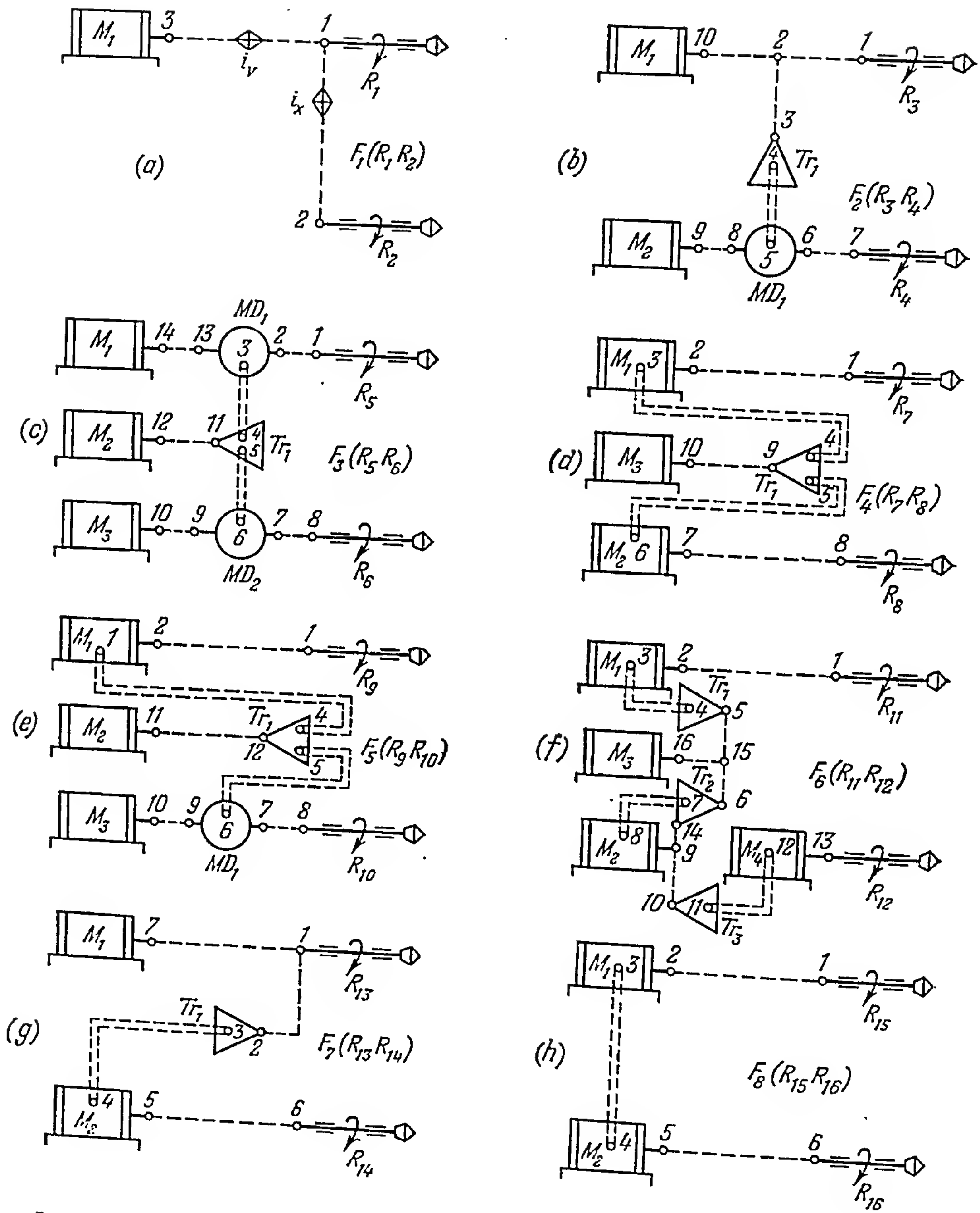

Fig. 5. Structures of complex kinematic groups producing two-element operative motion

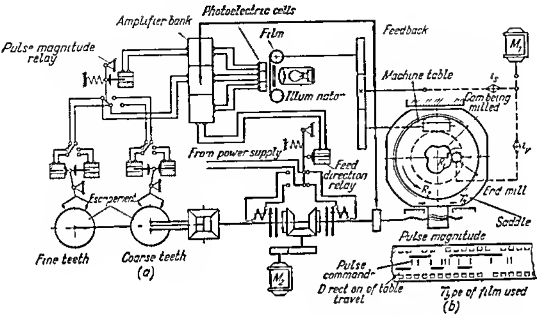

Fig 6 Structure of a complex kinematic group with motors arranged outside the internal kinematic constraint

external mechanical constraint provides the path length, velocity, direction and initial position of the operative motion

The structure of the kinematic group becomes more complex if the number of sources of motion or, more exactly, the number of initial motions in the group is more than one There may be several possible versions, in this case, of the structure of the group and arrangement of the sources of motion in reference to the internal constraint, as well as of the nature of this constraint

The kinematic group in Fig 5b, producing the operative motion $F_2 (R_3 R_4)$, has two sources of motion M_1 and M_2 and they are both outside the internal kinematic constraint The latter has several sections mechanical section 1-2 3, then transducer Tr_1 followed by power section 4 5 through which the power pulses are transmitted to the metering device, from which motion is transmitted to mechanical section 6 7

The initial motion originating in motor M_1 is fully used, that from motor M_2 is used only partly

An example of this type of structure is the feed motion group in the milling machine with a system of numerical controls devised by A Kobrinsky,

V. Besstrashnov and M. Breido (Fig. 6*a*). The cam blank is clamped on the rotary work table of this machine and its profile is machined with an end mill.

Two formative motions are produced in the machine: the cutting motion F_v (R_1), where R_1 is cutter rotation, and F_s (R_2T_3), where R_2 is table rotation and T_3 is radial travel of the table.

The structure of the first kinematic group producing the cutting motion F_v (R_1) is very simple and will not be considered here. We shall consider the structure of the operative group for producing the feed motion F_s (R_2T_3). The movable operative members of this group are the rotary table and its saddle. We shall investigate the internal kinematic constraint established between these members in the direction from the table to its saddle. This constraint consists of the work table, worm gearing, and three spur gears which drive the drum with the photographic film.

Three tracks of marks in the form of horizontal and vertical dashes are inscribed on the photographic film (Fig. 6*b*). Upon travel of the film a part of the light from the illuminator falls on the photoelectric cells (Fig. 6*a*). The electric signals originated in the cells pass through amplifiers and are transmitted to the corresponding relays. The pulse magnitude relay opens and closes the relays of the two escapements with fine and coarse teeth. Motion from motor M_2 is continuously being transmitted to the metering device in which the bodies of two magnetic clutches are freely mounted on the shaft of the saddle lead screw. The differential gearing is arranged at the left end of the saddle. When one of the magnetic clutches is engaged by means of the saddle direction relay, the lead screw will turn periodically through the angle permitted by the escapement that is switched into the constraint by the pulse magnitude relay. Elementary motion T_3—radial motion of the table saddle—is of the pulsed type.

The device, consisting of the illuminator, the drum with the film and the photoelectric cells, is the transducer in this case (see the diagram in Fig. 5*b*), while the mechanism with the escapements, differential gearing and magnetic clutches is the metering device. In addition to mechanical members, other devices, in which the kinematic constraint is accomplished by means of light, electricity and magnetism, are employed here in the internal kinematic constraint.

Figure 5*c* illustrates the structure of a kinematic group closely resembling that of the preceding group. The new group has three sources of motion and all are outside the internal constraint.

The second source of motion M_2 transmits motion to the transducer Tr_1 which, through sections *4-3* and *5-6* of the internal constraint, controls two metering devices. The latter take off a part of the motion from the first and third motors and transmit it to the movable operative members which, in our case, are spindles of the machine tool.

The parameters of elementary motions R_5 and R_6 are co ordinated by an internal constraint through mechanical section 1 2 and metering device MD_1, power section 3 4, transducer Tr_1, power section 5 6, metering device MD_2 and mechanical section 7 8 This arrangement ensures an entirely definite path of the operative motion F_3 (R_5R_6) An example of a structure of this type is the one used for the kinematic group of the feed motion in milling machines with electric-contact tracer control systems and, in particular, the electric contact tracer control system with automatic speed variation proposed by G Monakhov

A common feature of all the preceding structures of kinematic groups (Fig 5a, b and c) is that all the sources of motion are outside the internal kinematic constraint that determines the path of the operative motion

Shown in Fig 5d, e, f and g are structures in which the sources of motion are arranged both outside and inside the *internal kinematic constraint*

The group in Fig 5d has three sources of motion but only one is outside the internal kinematic constraint The initial motion is transmitted from this source to transducer Tr_1 Power pulses are transmitted from the latter through sections 4 3 and 5 6 of the constraint to motors M_1 and M_2 New initial motions appear on the output members of these motors and are transmitted through sections 2 1 and 7 8 of the constraint to produce the required elementary motions R_7 and R_8 on the spindles These elementary motions compose the formative motion F_4 (R_7R_8) Consequently, the operative constraint between motions R_7 and R_8 is accomplished through motors M_1 and M_2, i e through their power and mechanical parts This means that these motors are completely within the internal operative constraint, and any change in the parameters of their initial motions has an influence on the path of the operative motion F_4 (R_7R_8)

This structure has been realized in the numerically controlled milling machine, model 6H13np, incorporating the ENIMS control system A block diagram illustrating this system is shown in Fig 7a

The device for the numerical controls of a milling machine constitutes a separate kinematic group producing the complex feed motion F_s $(T_1T_2T_3)$ required in machining complex curvilinear surfaces such as is used in die sinking operations The feed motion F_s is composed of three elementary rectilinear motions longitudinal traverse T_1 of the table, cross traverse T_2 of the saddle, and vertical traverse T_3 of the milling head

As is the case with any kinematic group the numerical control device has an internal constraint consisting of two kinematic chains of the same type, interrelating the parameters of the elementary motions Thus, longitudinal traverse T_1 of the table interacts with cross traverse T_2 of the saddle through the following hydro electromechanical internal kinematic chain

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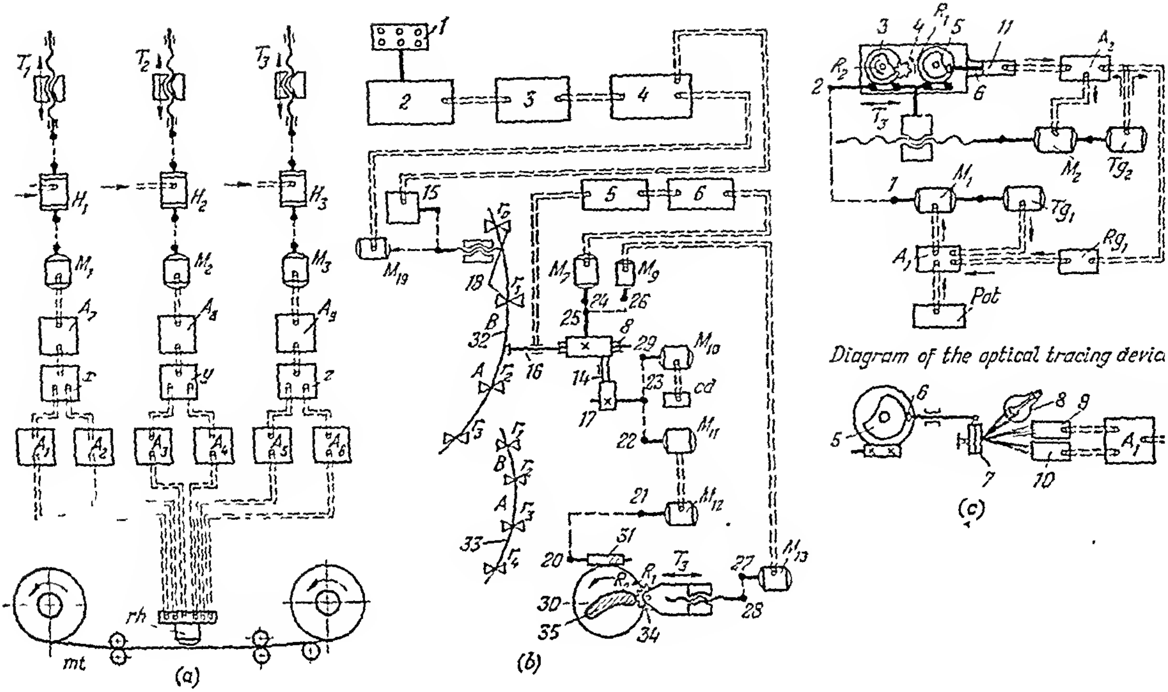

Fig. 7. Structural diagrams:

(*a*) numerically controlled milling machine (ENIMS): *mt*—magnetic tape; *rh*—reading head; A_1 through A_6—reading amplifiers; *x*, *y* and *z*—pulse distributing devices; A_7, A_8 and A_9—power amplifiers; M_1, M_2 and M_3—step motors; (*b*) numerically controlled lathe adapted for milling turbine blades: *1*—punched card; *2*—summation device; *3*—relay memory device; *4*—self-balanced bridge; *5*—spark device; *6*—amplifier; M_7—motor; *8*—rack-and-pinion drive; M_9—synchro-transmitter; M_{10} through M_{13}—motors; *14*—pusher; *15*—balancing potentiometer; *16*—stylus; *17*—cam; *18*—slides for moving the flexible template; M_{19}—motor; *20-21*, *22-23-29*, *24-26* and *27-28*—mechanical kinematic chains; *30*—worm wheel; *31*—worm; *32*—right-hand flexible template; *33*—left-hand flexible template; *34*—milling cutter; *35*—workpiece; *cd*—controlling device; r_0 through r_4—radii of the flexible template; (*c*) tracer-controlled profile milling machine: *1-2*—mechanical kinematic chain; *3*—workpiece; *4*—milling cutter; *5*—interchangeable template; *6*—stylus; *7*—mirror; *8*—lamp; *9* and *10*—photoelectric cells; *11*—optical tracing device; M_1 and M_2—motors; A_1 and A_2—amplifiers; Tg_1 and Tg_2—tachogenerators; Rg_1—regulator; *Pot*—potentiometer

ball-bearing lead screw of the table, hydraulic step motor H_1, electric step motor M_1, power amplifier A_7, device for distributing pulses along the x co-ordinate, reading amplifiers A_1 and A_2, magnetic tape reading head, magnetic tape, magnetic tape reading head, reading amplifiers A_3 and A_4, device for distributing pulses along the y co-ordinate, power amplifier A_8, electric step motor M_2, hydraulic step motor H_2 and the ball-bearing lead screw of the saddle. The second internal kinematic chain, between the saddle (motion T_2) and the milling head (motion T_3), has the same structure.

These chains operate as follows.

The tape feed with a separate three-phase induction motor feeds the magnetic tape (thickness 0 06 mm, width 19 2 mm) past the magnetic reading heads at a velocity of 100 mm per sec The length of the tape accommodated in the magazine (500 m) is sufficient for 1 5 hours of machine operation Recorded on the tape are six tracks of signals, two tracks for each elementary motion; one for each direction

The magnetic track consists of the record of consecutive pulses (6 to 8 pulses per mm) Each pulse turns the motors one step When the tape is fed uniformly at constant speed, the distance between successive pulses governs the rotational speed of the step motor Six magnetic heads (with clearances of 0 25 mm) read off the signals on the magnetic tape and the pulses are transmitted further to the reading amplifiers A_1 through A_6

The multistage semiconductor reading amplifiers have a total amplification of 5,000 to 7,000 The amplified signal is fed into devices for distributing the pulses between the x, y and z co-ordinates The purpose of these devices is to transmit the signals to the required section of the stator winding in the step servomotor to obtain the specified direction of its rotation From the pulse distributing device the signals are fed into power amplifiers A_7, A_8 and A_9 and then to the electric step motors The latter are reluctance pulse controlled motors of the inverted type or ones with an intermediate hollow rotor with various shaft rotation steps (1 5°, 3° or 6°) and with a maximum static torque of about 8 kgf-cm

The step servomotor turns the valve of the hydraulic step motor which has a high torque and is actually a hydraulic amplifier, or booster, with a torque amplification of 200 to 300 at an oil pressure of 15 to 20 kg per sq cm

The hydraulic motors drive the ball bearing lead screws When the hydraulic motor turns through an angular step equal to 3°, the nut of the ball-bearing lead screw travels a distance of 50 microns (0 05 mm)

In the internal constraint of the kinematic feed group under consideration there are three motors instead of two as shown in the diagram of Fig 5d This derives from the fact that a more complex operative motion is to be produced (three-element instead of two element motion)

The diagram in Fig 5e also has three motors but only one of them (M_1) is inside the internal operative constraint In this case a metering device is incorporated in the group

A more complex group is one with four sources of motion (Fig 5f) In this group, three motors, M_1 M_2 and M_4 are arranged within the internal constraint and the following internal constraint is established by the corresponding devices between the elementary motions B_{11} and B_{12} upper spindle, mechanical section *1 2*, motor M_1, power section *3-4* transducer Tr_1, mechanical section *5-6*, transducer Tr_2 power section *7-8* motor M_2, mechanical section *9 10*, transducer Tr_3 power section *11 12* of the constraint, motor M_4

and the lower spindle. The external constraint consists of motor M_3 and the mechanical section *16-15*.

An example of a machine tool with a kinematic group having this structure is a lathe adapted for milling turbine blades (Fig. 7*b*).

A compound arrangement of the sources of motion in a kinematic group is possible even where there are only two motors (Fig. 5*g*). Here the operative constraint between elementary motions R_{13} and R_{14} is accomplished through mechanical section *1-2*, transducer Tr_1, power section *3-4*, electric motor M_2 and mechanical section *5-6* of the constraint. Externally motor M_1 is linked with the operative constraint through the mechanical drive train *7-1*. In this group only one source of motion M_2 is within the internal constraint.

A structure of this type has been applied in the kinematic feed group of the tracer-controlled profile milling machine whose structural diagram is shown in Fig. 7*c*. Profiling (two-dimensional) operations are performed by this machine.

This profiler has two kinematic groups, one for the cutting motion F_v (R_1) and the other for the feed motion F_s (R_2T_3). The cutting motion group is simple and requires no further consideration. The feed motion group, on the other hand, is of the complex type. It has an internal constraint between the elementary motion R_2 (rotation of the circular table on which the work is clamped) and motion T_3 (longitudinal traverse of the table saddle).

A second rotary table, on which the interchangeable template is clamped, is also accommodated on the saddle. The servosystem is mounted on the saddle too (Fig. 7*c*).

As the template is rotated the stylus moves in a radial direction and turns the mirror. Arranged opposite the latter are two photoelectric cells which will be differently illuminated, depending upon the position of the mirror. This leads to a difference in photoelectric currents flowing to the amplifier and electric motor M_2 (Fig. 7*c*). If both photoelectric cells are illuminated with the same intensity, there will be no difference in the photoelectric currents and electric motor M_2 will stop. Thus, the internal constraint in this group is as follows: from the rotary table with the work through two worm gearing drives, and further from the template through the stylus and servosystem, amplifying device A_2 and electric motor M_2 to the saddle lead screw.

Motion R_2 is uniform and in one direction while motion T_3 is nonuniform and in alternating directions. This constraint provides the path of operative motion F_s (R_2T_3). The speed and other motion parameters are provided by the variable-speed electric motor M_1 controlled through a potentiometer and mechanical section *1-2* of the external kinematic constraint.

Besides these ordinary constraints, this group has an additional electrical constraint enabling the velocity parameter of operative motion to be varied

if such need arises In milling cams with sharp rises or drops in their profiles, it is necessary to reduce or increase the rate of feed to obtain the required shape This additional constraint automatically changes the rate of table rotation R_2 in accordance with the rate of traverse of the saddle T_3 without, however, altering the given ratio of the speeds of these motions at each moment in the course of time It is this ratio that determines the shape of the cam being milled To this end, the shaft of electric motor M_2 is linked rigidly to the shaft of the tachogenerator Tg_2 The latter transmits electric signals through amplifying device A_1 to electric motor M_1, changing its rotational speed relative to the mean rate of feed given through a potentiometer The shaft of electric motor M_1 is connected to tachogenerator Tg_1 which controls and maintains the given operational conditions of this motor

When this constraint is unnecessary, it is terminated by regulator Rg_1 Then the speed of electric motor M_1 is governed only by a single potentiometer

Thus, common to all kinematic groups having several sources of motion arranged both inside and outside the internal constraint, is the fact that the internal constraint producing the path of the operative motion passes through the power part of one or several electric motors Therefore the operation of these motors must meet the requirements made to internal constraints based upon the specified shape-accuracy indices of the surface being machined *Unforeseen variations in the speed of the output member of a motor arranged within the internal kinematic constraint* lead to a change in the path of the operative motion and a corresponding change in shape of the surface being machined

Figure 5*h* shows the structure of a kinematic group for the operative motion F_8 $(R_{15}R_{16})$ This group has two sources of motion and both are within the internal constraint of the group, i e the internal kinematic constraint passes through the power parts of these sources From the upper spindle the constraint is accomplished through mechanical section *1-2*, first electric motor M_1, power *section 3 4 second electric motor* M_2, and mechanical section *5 6* with the second spindle Here both motors influence the path of the operative motion produced by the group The velocity of this operative motion depends upon the speeds of both sources of motion if there are no special setting-up members in the mechanical sections *1-2* and *5 6* In this group the external constraint constituting the source of motion, coincides with the internal constraint

A structure of this kind can be found in the kinematic group of the cutting motion F_v (R_1R_2) in the Reishauer gear grinder model ZA (see Fig 70) which uses a helically profiled wheel

The structures of the kinematic groups illustrated in Fig 5 may be modified to some extent by a further development of the external and internal constraints, especially in cases when the parameters of the operative motion

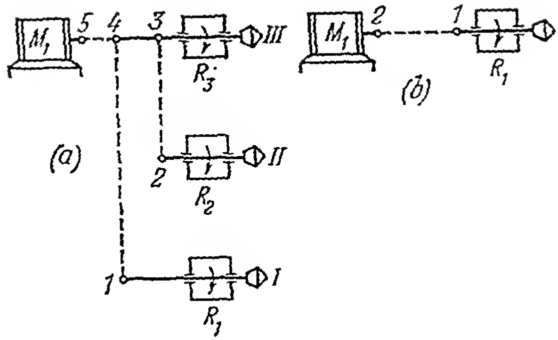

Fig. 8. Structures of kinematic groups for three- and single-element operative motions

are variable and governable. Essentially, these arrangements indicate the possible versions of structures for a kinematic group producing a complex two-element operative motion.

The structure of a group producing a complex three-element motion (Fig. 8*a*) can be modified by increasing the number of kinematic chains in the internal constraint. In producing a simple single-element operative motion (Fig. 8*b*), the internal constraint of the kinematic group is greatly simplified and consists of a single translatory or rotary kinematic pair. In this case, the motor can be arranged only in the external constraint.

A kinematic group can produce operative motions that differ in purpose. Therefore, the group is to be referred to by the same name as the operative motion it produces, i.e. formative (shaping), feed-in, indexing, controlling and handling motion groups.

CHAPTER 4

KINEMATIC STRUCTURE OF MACHINE TOOLS

4-1. Representative Kinematic Structures of the Formative Part of a Machine Tool

The kinematic structure of a machine tool may consist of a single kinematic group producing a single operative motion—the formative cutting motion In this case, the kinematic structure of the machine is that of the kinematic group Broaching machines can be mentioned as an example of such machine tools In them a single operative motion accomplishes several machining processes feed-in, shaping and, in some cases, even indexing

In most cases several operative motions are produced in a machine tool and its kinematic structure is composed of several kinematic groups Various combinations are feasible and therefore, the structure of a machine tool depends primarily upon the number of combined groups, their types and purposes

A machine tool may not necessarily have kinematic groups providing all the possible functional purposes Thus, for instance, machine tools frequently do not have an indexing group because the indexing process is not required in the operation of the machine tool, or this process is accomplished in conjunction with some other motion as for example, the formative motion The feed-in motion, as such, is also frequently absent The only kinematic groups without which no machine tool is conceivable are the formative, or shaping, groups It is these groups that determine the kinematic structure of the machine tool The other kinematic groups, accomplishing the indexing and feed in processes may modify the kinematic structure of the machine tool to a greater or lesser extent but do not determine it. Of equal influence on the structure of the machine tool is the nature of the kinematic interconnections between the groups through internal and/or external kinematic constraints

The structure of the machine tool may also be influenced by such factors as the shape of the cutting tool, arrangement of the basic units of the machine the level of operating accuracy required, etc

Consequently, the structure of a machine tool implies primarily the number and composition of the formative kinematic groups and their intergroup constraints, and then the number and composition of the remaining kinematic groups, having other functions and the remaining kinematic intergroup constraints

Diversified as it may be, any kinematic structure found in metal-cutting machine tools belongs to one of three classes of kinematic structures consisting of the following formative kinematic groups:

(a) only simple groups producing a single-element operative motion; the path of each motion being produced by an internal constraint in the form of rotary or rectilinear translatory kinematic pairs; such a structure being denoted by the letter E and called an *elementary structure*;

(b) only complex groups, each producing an operative motion that consists of two or more elementary motions; the internal constraint of such groups consisting of one or several kinematic chains; such a structure being denoted by the letter C and called a *complex structure*;

(c) both simple and complex groups; such a structure being denoted by the letter K and called a *combined structure*.

In the designation of a structure, the letter E, C or K is followed by a figure indicating the number of formative kinematic groups composing the structure, and a second figure indicating the total number of elementary motions making up all the formative operative motions.

Depending upon the number and complexity of the formative operative motions produced, various versions of kinematic structure are feasible in each group (Fig. 9*A*).

Figure 9*B* gives examples illustrating the kinematic structures of various machine tools having representative structures. Thus, machine tools with an elementary structure (class E) are shown in Fig. 9*a*, *b* and *c*.

A broaching machine is shown schematically in Fig. 9*a*. It has one simple formative kinematic group producing a simple rectilinear cutting-speed motion which accomplishes three processes: shaping, feed-in and, in some cases, indexing. Its structure designation is E11.

The lathe for turning the races in ball-bearing rings, shown in Fig. 9*b*, consists of two simple formative groups according to the structure class E22.

A cylindrical grinder with a structure E33 is shown in Fig. 9*c*. Each kinematic group in this type of machine has its own electric or hydraulic motor.

The diagrams of Fig. 9*d*, *e* and *f* represent machine tools with a complex structure, class C. but having a single complex formative group. The degree of complexity of this group may vary. Diagrams of three machine tools are illustrated. These are: an engine lathe for cutting scrolls and having a structure of class C12 (Fig. 9*d*) which produces a complex two-element formative motion; an engine lathe for cutting taper threads with a structure class C13 (Fig. 9*e*) in which the motion is of the three-element type; and a relieving lathe for taper taps with a structure of class C14 (Fig. 9*f*). The structure of these machine tools becomes more complex with the number of elementary motions due to the more complex internal constraint. Thus, the internal constraint in the first machine consists of a single kinematic chain with the

A Representative Kinematic Structures of Machine Tools

Number of groups		1				2					3			
Number of elementary motions		1	2	3	4	2	3	4	5	6	3	4	5	6
Classes	E	E11				E22					E33			
	C		C12	C13	C14			C24	C25	C26				C36
	K						K23	K24	K25	K26		K34	K35	K36

B Examples of Machine Tools with Representative Kinematic Structures

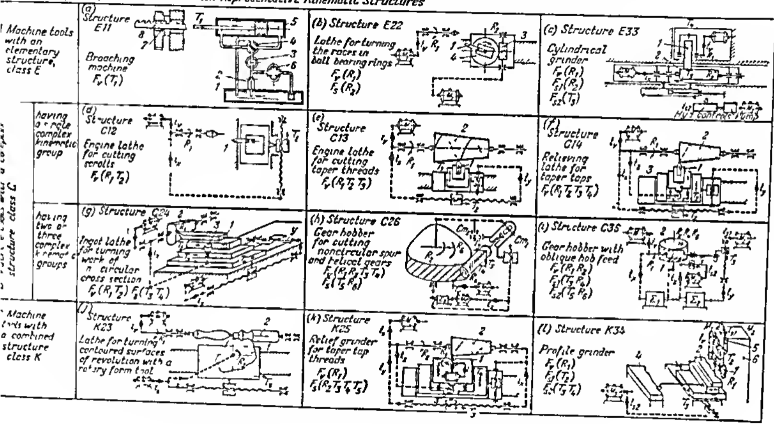

Fig 9 Classification of kinematic structures used in metal cutting machine tools

change-gear quadrant i_x while that of the third machine consists of three kinematic chains with the change-gear quadrants i_x, i_y and i_z.

The diagrams of Fig. 9*g*, *h* and *i* also illustrate machine tools with complex structures, class C, but having two or three complex formative groups.

The diagram in Fig. 9*g* is for an ingot lathe which turns tapered ingots of noncircular cross section. Its structure is of class C24 and consists of two complex formative groups. The structure has been simplified by the provision of a separate motor in each group.

The gear hobber for cutting noncircular spur and helical gears (Fig. 9*h*) has a class C26 structure. The cutting-motion group in this arrangement produces a very complex motion composed of four elementary motions. The internal constraint in this case is especially complicated.

The gear hobber with oblique hob feed has a class C36 structure (Fig. 9*i*). Here there are already three complex kinematic groups and the table with the gear blank is included in all three groups. To this end, the machine has two summation mechanisms $\sum_1$ and $\sum_2$ (differentials). The arrangement is even more complex than that of the preceding machine tools.

Figure 9*j*, *k* and *l* illustrates diagrams of machine tools with combined structures K. They include a lathe for turning contoured surfaces of revolution with a rotary form tool, structure class K23 (Fig. 9*j*); a machine for relief grinding the thread of a taper tap, structure class K25 (Fig. 9*k*); and a profile grinder for templates, class K34 (Fig. 9*l*). Typical of these machines is that they always have at least two formative kinematic groups, and one of them is sure to be a simple group.

This classification of the kinematic structures of machine tools according to the kind and number of formative groups enables machine tools differing in their processing features to be systematized in accordance with their kinematic data. Thus, all the structures can be reduced to 12 existing machine tools with representative structures whose investigation offers no great difficulties.

The structure of the principal part of a machine tool provides for the shaping, indexing and feed-in processes and therefore it consists of several kinematic groups, including the groups for the cutting speed and feed, feed-in and indexing motions. The structure of the principal part of the machine is obtained as the combination of a series of complex and diversified kinematic and constructional interconnections of these groups and the use of operative members and motors common to several groups.

The structure of the principal part of the machine tool depends, not only upon the number of groups and their types, but also upon the kinematic interconnection of the kinematic groups that has been applied in the given machine tool. Therefore, using various methods of interconnecting the kinematic groups, several different versions of the structure of the formative part can be obtained for each type of machine tool.

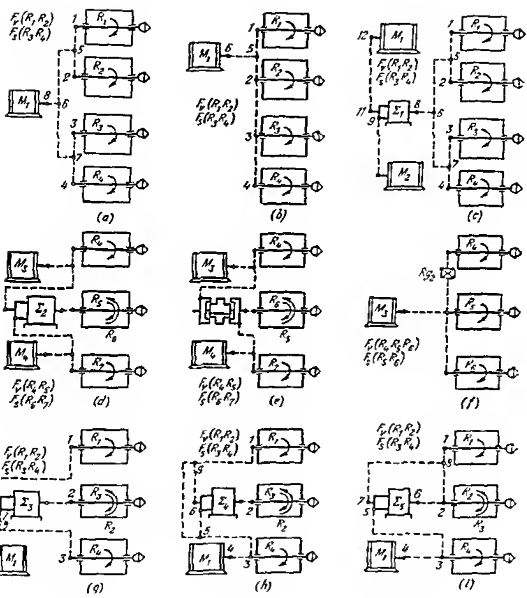

Fig 10 Methods of interconnecting kinematic groups

Let us consider the factors governing the selection of the method for interconnecting the kinematic groups and, consequently, the selection of the structure for the formative part of the machine tool.

1. First of all, it is necessary to decide whether or not the structure of the machine tool has a common movable operative member, i.e. whether there is a movable operative member that belongs to two kinematic groups.

If there is no such member, then the groups cannot be interconnected through an internal constraint, since the groups are independent, but they can be connected through external constraints. In this case, the structure of the formative part of the machine tool depends upon the number of drive motors. If there is only one motor and it is common to all the groups, two cases are feasible.

In the first case, the connected group is driven from the external constraint of the first group (Fig. 10*a*) and the external constraint of the two groups is through the common interlinking section *8-6*.

In the second case (Fig. 10*b*), the connected group is driven directly from some movable operative member of the first group or from some other member, but only one within the internal constraint of the first group. Then the external constraint of the second group will consist of sections *6-5-2-3*. If the arrangement has two motors in common (Fig. 10*c*), the kinematic interconnection of the groups along external constraints is possible only through a summation mechanism (differential).

If the group being connected has its own (separate) motor, then the groups cannot be interconnected if there is no operative member in common. Only a constructional interconnection is possible.

2. If the machine tool has a movable operative member in common, the method of interconnecting the groups depends upon the action of the operative motions in the course of time. These actions may be either simultaneous or occur at different times.

With simultaneous motions, when the common movable member participates at the same time in two kinematic groups, the groups are interconnected through a differential (Fig. 10*d*). Here, the operative member in common and the differential interconnect both groups through internal constraints. This type of structure is used in gear hobbing machines. The work spindle belongs to two groups—cutting speed and feed—simultaneously. When such a machine tool has one or several motors, the groups can be interconnected along external constraints, by the procedures indicated above (Fig. 10*a* and *b*).

If the operative motions produced, for example, by two separate kinematic groups, are not to be simultaneous, the kinematic groups are also interconnected along internal constraints, through a common movable operative member, but the method of interconnection will be different. In this case, the group interconnection may be *parallel*, *series* or *series-parallel*.

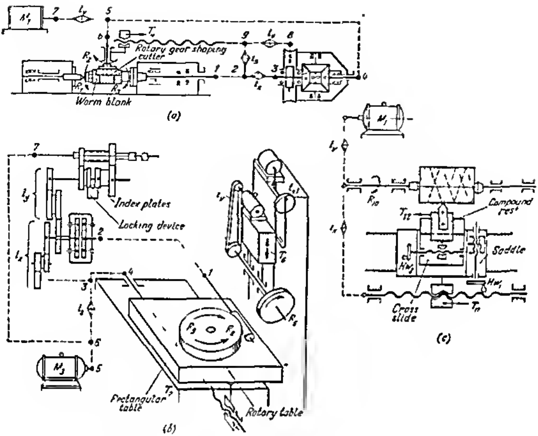

Fig. 11. Diagrams of machine tools with parallel group interconnection
(a) model E3-10A worm thread generator for cutting multiple-start worms (b) model 5831 gear grinder (c) engine lathe

Parallel interconnection of groups means that a common operative member, in which the elementary motions are combined, can simultaneously perform both motions in parallel, even when the durations of the motions differ.

model E3-10A worm thread generator for cutting multiple-start worms with a rotary cutter (gear shaping cutter). Its structure diagram is shown in Fig. 11*a*. In this machine tool, two complex groups, producing the opera-

tive motions F_v (R_1R_2) and F_s (R_3T_4) are kinematically interconnected. The internal kinematic constraints of the two groups are through the summation mechanism (bevel-gear differential). The work spindle belongs to both kinematic groups. It is connected to the cutter spindle through the internal kinematic chain: $1 \to 2 \to i_x \to 3 \to$ central gears of the differential and mechanical sections *4-5-6* of the constraint. The work spindle is also connected to the lead screw through the kinematic chain: $1 \to 2 \to i_x \to 3 \to$ left central gear of the differential→planet gears and worm gearing, and section $8\text{-}i_y\text{-}9$ of the constraint.

Motion is transmitted from the motor to the internal chain of the first group through the drive train $7\text{-}i_v\text{-}5$. The drive train of the second group for the feed motion F_s (R_3T_4) is somewhat longer and parts of the internal chains of the first group are used to transmit motion. Motion is transmitted from the motor to the internal constraint through sections: $7\text{-}i_v\text{-}5\text{-}4\text{-}3\text{-}i_x\text{-}2\text{-}i_s\text{-}9$.

Two complex formative groups are interconnected in this machine. Simple and complex groups can be interconnected in the same way. This has been used in the gear grinder, model 5831 (Fig. 11*b*), in which the indexing group (with change-gear quadrant i_y) is interconnected with the profiling group (with change-gear quadrant i_x) in parallel through a differential.

Indexing is sometimes effected in this manner in an engine lathe in cutting multiple-start threads (Fig. 11*c*). To this end the compound rest with the toolpost is set so that its hand traverse, motion T_{12}, is parallel to motion T_{11} of the saddle. After cutting the first start, the compound rest is traversed by hand a distance equal to the pitch, the latter being a fraction of the lead. This distance equals $l = \frac{P}{k}$ where k is the number of starts of the thread and P is its lead. The summation mechanism in the given case is the compound rest which enables the toolpost to be traversed without disengaging the motion of the saddle.

In series interconnection of groups (see Fig. 10*e*), the common operative member participates alternately in one or the other operative motion. A distinguishing feature of a machine tool structure in which the kinematic groups are interconnected in series is the provision of a mechanism which disengages and re-engages an internal or external kinematic chain. This mechanism may be a claw clutch (single-tooth), differential, indexing plate or other device. Interconnection in series is most often applied in interconnecting a complex group with a simple one as, for instance, the formative and indexing groups.

The thread-milling machine, model 561 (Fig. 12*a*), is an example of group interconnection in series. Here, the helical motion group F_s (R_1T_2) is connected in series with the indexing motion group *Ind* (R_3). The latter is accomplished by hand, by turning the handle of the locking member on the index

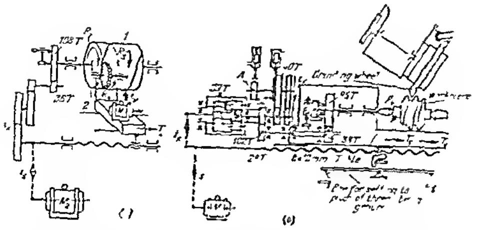

Fig 12 Diagrams of machine tools with series group interconnection
(a) model 561 thread milling machine (b) thread grinding machine model 5883A for grinding worms

plate The plate has 108 equally spaced holes in its end face When the locking member is retracted from a hole of the index plate, the thread-cutting gear train between the work and lead screw is disengaged, motion $F_s (R_1 T_2)$ ceases, and the indexing motion is imparted to the work by turning the indexing handle

The same feature is found in the thread grinding machine, model 5883A (Fig 12b), except that the differential plays the part of the mechanism for disengaging the thread cutting gear train In this particular case, the differential operates, not as a summation device but as a clutch Of its three members, only two are in operation at any time

Compound (*series parallel*) *group interconnection* (see Fig 10f) is based on the resolution of one complex motion into two less complex ones This *interconnection is most frequently applied in combining a complex formative group* with a simple indexing group For this purpose, a special reversing device (not subject to relative slip of the members at the moment of reversal) *is installed* within the internal constraint of the complex group Upon the reversal of the elementary motion of the common operative member the complex formative motion is terminated and only the simple indexing motion of this common operative member remains Thus, at first there are two processes—shaping and indexing—and then only indexing

Such a special reversing device in the *form of a* composite gear, has been used in the gear grinder, model 5П84 whose simplified kinematic diagram is shown in Fig 13c

tive motions $F_v (R_1R_2)$ and $F_s (R_3T_4)$ are kinematically interconnected. The internal kinematic constraints of the two groups are through the summation mechanism (bevel-gear differential). The work spindle belongs to both kinematic groups. It is connected to the cutter spindle through the internal kinematic chain: $1 \rightarrow 2 \rightarrow i_x \rightarrow 3 \rightarrow$ central gears of the differential and mechanical sections *4-5-6* of the constraint. The work spindle is also connected to the lead screw through the kinematic chain: $1 \rightarrow 2 \rightarrow i_x \rightarrow 3 \rightarrow$ left central gear of the differential→planet gears and worm gearing, and section $8\text{-}i_y\text{-}9$ of the constraint.

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In series interconnection of groups (see Fig. 10*e*), the common operative member participates alternately in one or the other operative motion. A distinguishing feature of a machine tool structure in which the kinematic groups are interconnected in series is the provision of a mechanism which disengages and re-engages an internal or external kinematic chain. This mechanism may be a claw clutch (single-tooth), differential, indexing plate or other device. Interconnection in series is most often applied in interconnecting a complex group with a simple one as, for instance, the formative and indexing groups.

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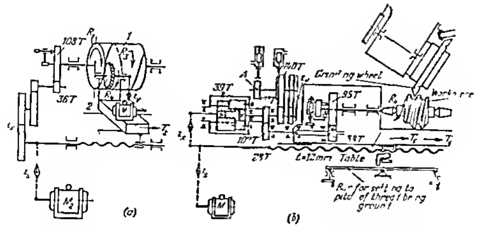

Fig. 12. Diagrams of machine tools with series group interconnection
(a) model 561 thread milling machine (b) thread grinding machine, model 5883A, for grinding worms

plate. The plate has 108 equally spaced holes in its end face. When the locking member is retracted from a hole of the index plate, the thread-cutting gear train between the work and lead screw is disengaged, motion F_s (R_1T_2) ceases, and the indexing motion is imparted to the work by turning the indexing handle.

The same feature is found in the thread grinding machine, model 5883A (Fig. 12b), except that the differential plays the part of the mechanism for disengaging the thread-cutting gear train. In this particular case, the differential operates, not as a summation device, but as a clutch. Of its three members, only two are in operation at any time.

Compound (*series parallel*) *group interconnection* (see Fig. 10f) is based on the resolution of one complex motion into two less complex ones. This interconnection is most frequently applied in combining a complex formative group with a simple indexing group. For this purpose, a special reversing device (not subject to relative slip of the members at the moment of reversal) is installed within the internal constraint of the complex group. Upon the reversal of the elementary motion of the common operative member, the complex formative motion is terminated, and only the simple indexing motion of this common operative member remains. Thus, at first there are two processes—shaping and indexing—and then only indexing.

Such a special reversing device, in the form of a composite gear, has been used in the gear grinder, model 5П84, whose simplified kinematic diagram is shown in Fig. 13c.

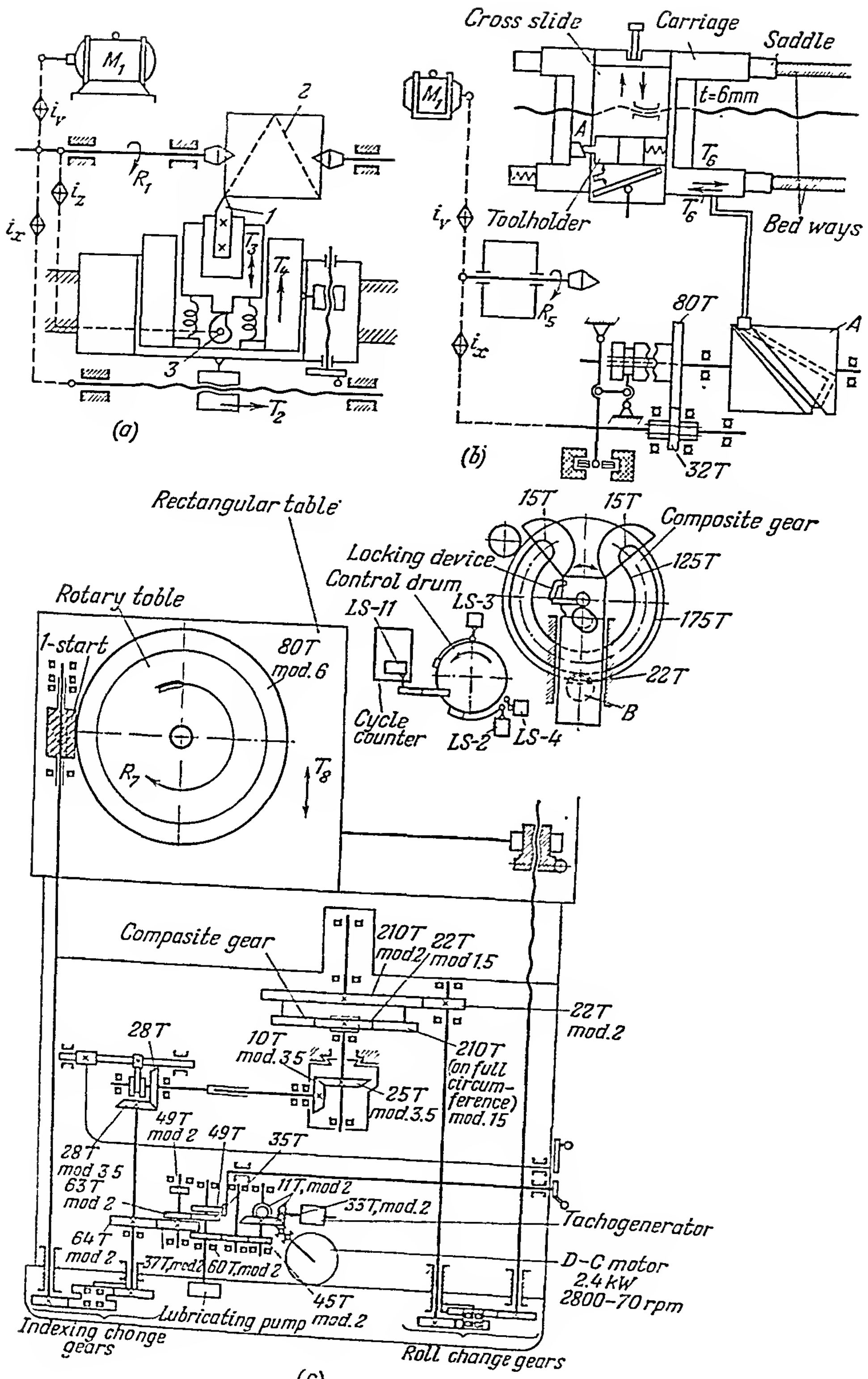

Fig. 13. Diagrams of machine tools with series-parallel group interconnection: (a) relieving lathe; (b) threading lathe, model 1921; (c) gear grinder, model 5П84

This composite gear is reversed by means of the travelling drive pinion B, which alternately meshes with the external and internal teeth of the composite gear. The rectangular table travels alternately in both directions (motion T_8) while the circular table continues to rotate only in one direction (motion R_7). As a result, the workpiece executes the roll (generating) motion F_s (R_7T_8) during the forward stroke of the rectangular table. The motion F_s stops upon table reversal while the indexing motion Ind (R_7) of the circular table continues.

Similar interconnection of the formative and indexing groups is typical of all relieving lathes (Fig. 13a). Here a plate cam 3 is used as the special reversing device within the internal constraint of the formative group.

A compound method of indexing is applied in the threading lathe, model 1921 (Fig. 13b) for cutting multiple-start threads. Upon the return of the tool to its initial position, after the first start, the spindle continues to rotate in the same direction, not a whole number of revolutions, but a certain fraction of a revolution more or less, enabling the tool to register in the next start. Cylinder cam A, installed in place of an ordinary lead screw, is the special reversing device in this case.

These general theoretical propositions concerning the kinematic structure of the formative part of machine tools are best understood in analysing the structures of machine tools of a single processing group, for example, gear hobbing machines.

Figure 14 illustrates the construction of the representative structures employed for the formative part of a gear hobbing machine in performing the most complex type of machining, i.e. in cutting a helical gear with a hob.

A diagram of the formative motion is given in Fig. 14a. Two formative motions are required to cut a helical gear with a hob. The first of these, the cutting speed motion F_v (R_1R_2) shapes the tooth profile and also provides for the indexing process. This is called the roll, or generating, motion and is produced by two elementary motions: hob rotation R_1 and table rotation with the gear blank R_2.

The second, feed motion F_s (T_3R_4) shapes the tooth along its length, along a helix. This helical motion is produced by two elementary motions: rectilinear motion of the hob across the gear face T_3 and rotation of the gear blank R_4.

Both of the formative motions are of the complex two-element type and to produce them the formative part of the gear hobber must be designed in the form of two complex formative kinematic groups: the cutting motion F_v (R_1R_2) and the feed motion F_s (T_3R_4). In addition to these two formative groups, the formative part of the hobber may incorporate a third group—the feed-in group. The composition of the latter group depends upon the feed-in method applied in the hobber. An analysis of the structure of the formative part of a gear hobber with a separate feed-in group is given in Sec. 5.3.

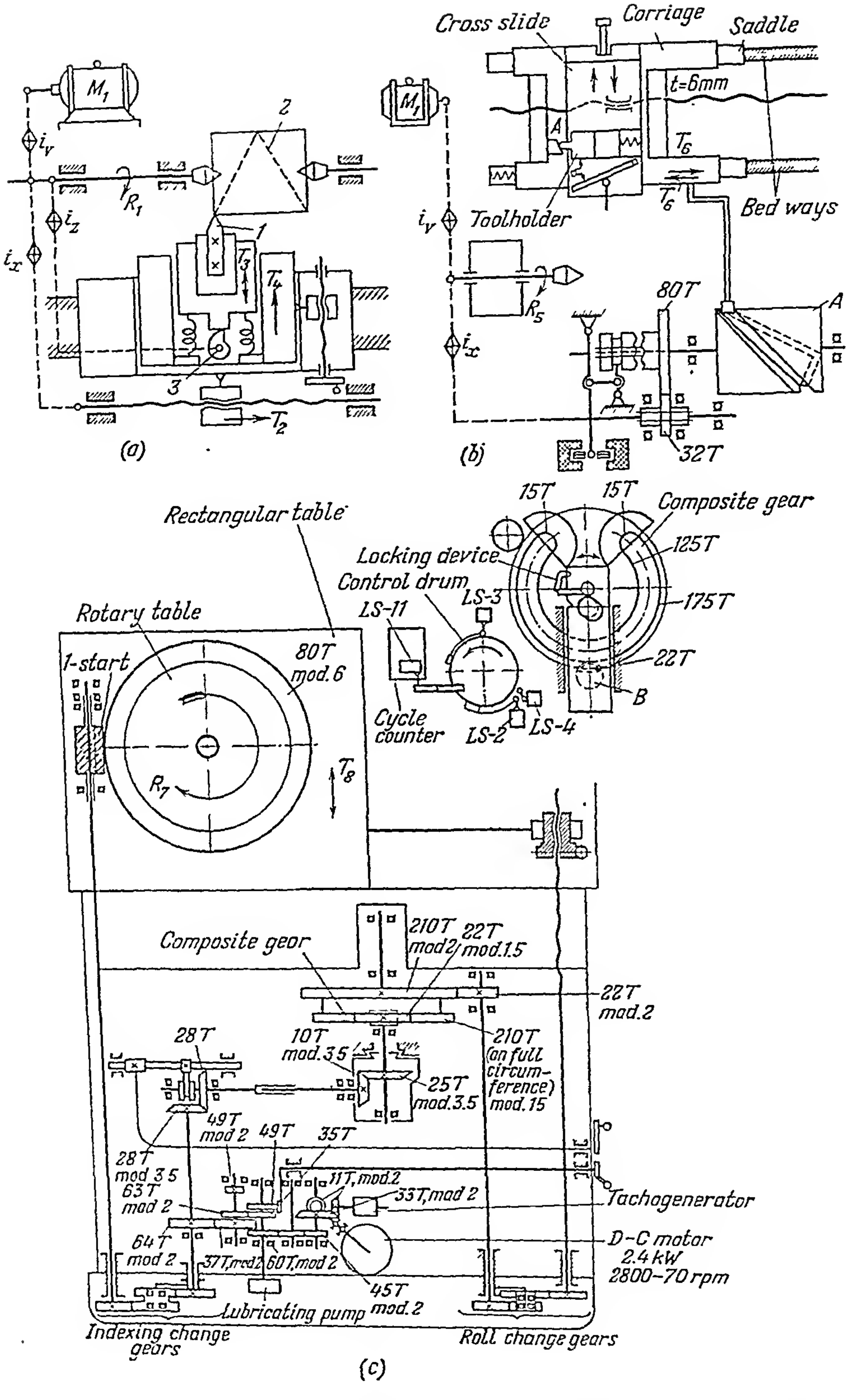

Fig. 13. Diagrams of machine tools with series-parallel group interconnection: (a) relieving lathe; (b) threading lathe, model 1921; (c) gear grinder, model 5П84

This composite gear is reversed by means of the travelling drive pinion B, which alternately meshes with the external and internal teeth of the composite gear The rectangular table travels alternately in both directions (motion T_8) while the circular table continues to rotate only in one direction (motion R_7) As a result, the workpiece executes the roll (generating) motion F_s (R_7T_8) during the forward stroke of the rectangular table The motion F_s stops upon table reversal while the indexing motion Ind (R_7) of the circular table continues

Similar interconnection of the formative and indexing groups is typical of all relieving lathes (Fig 13a). Here a plate cam 3 is used as the special reversing device within the internal constraint of the formative group

A compound method of indexing is applied in the threading lathe, model 1921 (Fig 13b) for cutting multiple start threads Upon the return of the tool to its initial position, after the first start, the spindle continues to rotate in the same direction, not a whole number of revolutions, but a certain fraction of a revolution more or less, enabling the tool to register in the next start Cylinder cam A, installed in place of an ordinary lead screw, is the special reversing device in this case

These general theoretical propositions concerning the kinematic structure of the formative part of machine tools are best understood in analysing the structures of machine tools of a single processing group, for example, gear hobbing machines

Figure 14 illustrates the construction of the representative structures employed for the formative part of a gear-hobbing machine in performing the most complex

A diagram of

motions are requ

cutting-speed motion F_v (R_1R_2), shapes the tooth profile and also provides for the indexing process This is called the roll, or generating, motion and is produced by two elementary motions hob rotation R_1 and table rotation with the gear blank R_2

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Both of the formative motions are of the complex, two element type and to produce them the formative part of the gear hobber must be designed in the form of two complex formative kinematic groups the cutting motion F_v (R_1R_2) and the feed motion F_s (T_3R_4) In addition to these two formative groups, the formative part of the hobber may incorporate a third group—the feed in group The composition of the latter group depends upon the feed in method applied in the hobber An analysis of the structure of the formative part of a gear hobber with a separate feed in group is given in Sec 5 3

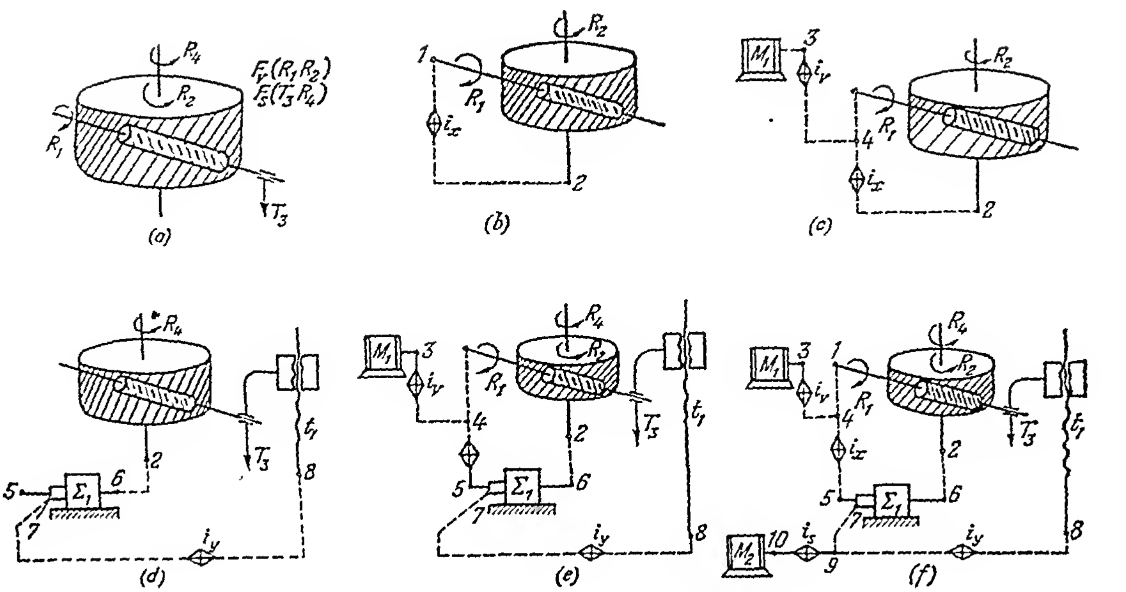


Fig. 14. Construction of the representative structure employed for the formative part of a gear-hobbing machine

The whole formative part of the hobber is constructed by evolving the first formative kinematic group, the cutting-speed motion group F_v (R_1R_2). It is known that any kinematic group always consists of two kinematic constraints: the internal constraint which provides the path of the given motion, and the external constraint which provides the velocity, path length, direction and initial point of the given motion. First of all, we construct the internal constraint of the kinematic group for the cutting-speed motion, Fig. 14*b*. To this end we connect the hob spindle to the work table through a rigid (positive) kinematic chain (a dash line in the diagram) which makes the table rotate with the corresponding speed upon any rotation of the hob spindle. Thus, through the internal kinematic constraint $R_1 \to 1 \to i_x \to 2 \to$ $\to R_2$, the elementary motions R_1 and R_2 become interconnected and thereby produce the path of the generating motion.

If this internal kinematic chain is connected to the motor (Fig. 14*c*), then kinematic chain $M_1 \to 3 \to i_v \to 4$ will be the external kinematic constraint in which point *4*, the point of connection of the external and internal constraints, is the member which belongs to both the external and internal constraints. These two constraints produce the complex generating motion which is also the cutting motion F_v (R_1R_2). The structure of this kinematic

group can be set down in accordance with the structural diagram as follows

$$\begin{tikzcd}
 & & & & 1 \arrow[r] & R_1 \text{ of the hob} \\
M_1 \arrow[r] & 3 \arrow[r] & i_v \arrow[r] & 4 \arrow[ur] \arrow[dr] & & \\
 & & & & i_x \arrow[d] & \\
 & & & & 2 \arrow[r] & R_2 \text{ of the work table}
\end{tikzcd}$$

where i_v = cutting speed change gears
i_x = indexing change gears

It is evident from this conventional schematic representation that the motion is transmitted from motor M_1 through the external constraint to point *4*. It is transmitted further, upward to the hob and simultaneously downward to the work table.

The second kinematic group, the feed motion group F_s (T_3R_4), can also be constructed separately. The construction is begun with the internal constraint (Fig. 14*d*). In this case—one frequently encountered in machine tools—one movable operative member (the work table) participates in producing both formative motions. Hence, the table belongs to both groups and receives two independent elementary motions R_2 and R_4. To this end, a summation mechanism is included in the arrangement. This mechanism $\sum_1$ (differential) has two input members (points *5* and *7*) and one output member (point *6*) to which the sum or difference of the two elementary motions R_2 and R_4 is imparted. The internal constraint of the group is called the *differential train*, it interconnects the vertical feed screw with the table through the constraint

$$T_3 \rightarrow t_1 \rightarrow 8 \rightarrow i_y \rightarrow 7 \rightarrow \textstyle\sum_1 \rightarrow 6 \rightarrow 2 \rightarrow R_4$$

where t_1 = pitch of the vertical feed screw
$\sum_1$ = differential.

If the previously constructed cutting speed motion group F_v (R_1R_2) is connected to this internal constraint, we obtain the intermediate diagram shown in Fig. 14*e*. It is now evident that the internal constraints of both groups pass through differential $\sum_1$. To construct the first, and more simple, version of the kinematic arrangement of a gear hobber, we shall take the case in which the feed group has its own separate motor M_2.

The structure of the whole feed group F_s (T_3R_4) can then be set down as (Fig. 14*f*)

$$\begin{tikzcd}
 & & & 7 \arrow[r] & \sum_1 \arrow[r] & 6 \arrow[r] & 2 \arrow[r] & R_4 \\
M_2 \arrow[r] & 10 \arrow[r] & i_s \arrow[r] & 9 \arrow[u] \arrow[d] & & & & \\
 & & & i_y \arrow[r] & 8 \arrow[r] & t_1 \arrow[r] & T_3 &
\end{tikzcd}$$

It follows from this representation that motion is transmitted from motor M_2 through the external constraint up to point *9*. Further, from this point, motion is transmitted upward through the differential to the work table, and simultaneously downward through the differential change gears i_y and vertical feed screw to the hob slide.

Connecting the external constraint of the feed group to the whole previous arrangement, we obtain the structure of the formative part of a gear hobber in which each formative group has its own separate motor (Fig. 14*f*).

This structure has been employed in the gear hobber, model 5312, which was built into an automatic transfer machine for gear production developed by ENIMS (Experimental Research Institute for Metal-Cutting Machine Tools, Moscow).

This proves to be the simplest and most efficient version of gear hobber kinematic structure as witnessed by its excellent performance in the operation of the transfer machine.

However, the main drawback of this version must also be mentioned: only feed per minute can be obtained, feed per revolution of the work (which we shall call simply "feed per revolution") is not available, which fact distinguishes the given machine from other gear hobbers. This shortcoming becomes especially important if the cutting-speed change gears must be frequently changed over, because it entails a corresponding change-over in the feed change gears to maintain the optimum feed per revolution. Such additional change-over complicates the setting-up of the hobber. It should also be mentioned that a hobber with two motors is somewhat more expensive. Thus, even though gear hobbers with two motors have not found wide application in general practice, their use in the above-mentioned transfer machine was quite justified and expedient.

Other versions can be developed for the structure of the formative part of a gear hobber if the constitution of the external constraint is varied in the feed group. Let us consider these versions, which are illustrated in Fig. 15.

It is possible to maintain constant feed per revolution in gear hobbers in which each formative group has its separate motor if a special intergroup kinematic constraint is provided in the structural arrangement between the formative groups. This intergroup constraint maintains constant feed per revolution (Fig. 15*a*) through the chain: work table $\rightarrow 2 \rightarrow 8 \rightarrow 9$. Oil from an ordinary pump is delivered through a metering pump (*MP*) to hydraulic cylinder C_1. The faster the table rotates, the larger the volume of oil discharged by the metering pump and the higher the feed per minute of the hob slide. This provides for a constant feed per revolution. This structural arrangement has been employed in the Reishauer, model ZA, gear grinder, operating with a helically profiled grinding wheel (Fig. 70). The addition of this short intergroup constraint did not complicate the kinematics of the machine to any great extent, but the constraint ensures a more uniform

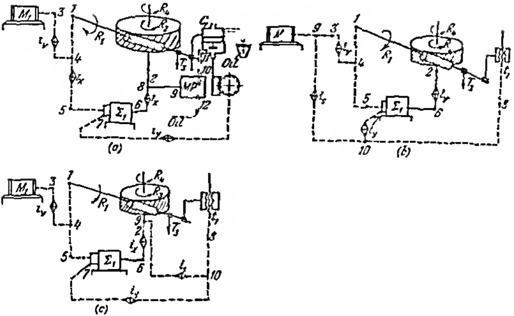


Fig. 15. Versions of the external constraints in the kinematic feed group of gear hobbers

wheel feed and, consequently, a better surface finish on the ground teeth. The same principle is to be used in gear hobbing machines.

Machine tools are commonly built with one drive motor to reduce their cost (Fig. 15*b* and *c*). Two different structural arrangements may be applied in this case. The essence of the first arrangement (Fig. 15*b*) is that the external constraint of the feed group F_s (T_3R_4) is connected to the external constraint of the cutting speed motion group F_v (R_1R_2) at point *9*, located near motor M_1. This arrangement has not been used in gear hobbers because the drive kinematic chains are longer than in the preceding versions and only feed per minute is available. If, on the other hand, the external constraint of the feed motion group F_s (T_3R_4) is connected to the internal constraint of the cutting speed motion group F_v (R_1R_4) at point *9* located near the work table (Fig. 15*c*), the kinematic arrangement of the machine becomes more compact and it provides for constant feed per revolution when the cutting speed is changed. For these reasons this structure is employed in almost all gear hobbing machines and it can be accepted as the representative structure for this class of machine. Nevertheless, the structure of the kinematic feed group is quite cumbersome and more involved than in the preceding version, though it has not made the whole structure more com-

plex. This is due to the fact that the external and part of the internal constraint of the first kinematic group for the cutting-speed motion are used as the external constraint of the second, feed motion, group.

To establish the constitution of the feed group, we remove the cutting-speed motion group from the general structure of the formative part of the gear hobber (Fig. 15*c*). The remaining part of the structure refers to the feed motion group $F_s\,(T_3R_4)$. Representing this structure conventionally we obtain

$$
\begin{tikzcd}
 & & & & R_4 & 9 \arrow[l] \arrow[r] & i_s \arrow[r] & 10 \arrow[r] \arrow[dd] & 8 \arrow[r] & t_1 \\
 & & & & & 2 \arrow[u] & & & & \\
M^1 \arrow[r] & 3 \arrow[r] & i_v \arrow[r] & 4 \arrow[d] & & i_x \arrow[u] & & i_y \arrow[dd, dash] & & \\
 & & & 5 \arrow[r] & \Sigma_1 \arrow[r] & 6 \arrow[u] & & & & \\
 & & & & 7 \arrow[u] & & & {} \arrow[lll] & &
\end{tikzcd}
$$

This representation shows that motion is transmitted from drive motor M_1 through the external constraint of the feed group, speed change gears i_v, differential Σ_1, indexing change gears i_x, point *9*, and feed change gears i_s to point *10* which is the interconnection of the external and internal constraints in the feed group $F_s\,(T_3R_4)$. From point *10* motion is transmitted along the internal constraint of the feed group through point *8* and vertical feed screw to the hob slide which accomplishes the elementary motion T_3. Motion is likewise transmitted from point *10* through the differential change gears i_y, second input shaft *7* of the differential, differential Σ_1, output shaft *6* of the differential, indexing change gears i_x and point *2* to the work table to which the second elementary motion R_4 is imparted. This group produces the helical feed motion $F_s\,(T_3R_4)$. Since the feed change gears i_s are arranged following the differential and the indexing change gears i_x, the hobber has a feed per revolution which remains constant upon any change-overs of change gears i_x and i_y, if change gears i_s are not altered. The feed per minute depends upon setups of the change gears i_v, i_x and i_s.

This representative structure of the formative part (Fig. 15*c*) is applied in the most widely used gear hobbers in the Soviet Union (models 5Д32 and 5E32 of the Komsomolets Plant) as well as in many other Soviet and foreign models.

It is also evident from the structural diagram that the external constraint of the feed group, between motor M_1 and point *10*, is not an independent and separate feed gear train, but consists of a series of sections belonging to the cutting-speed motion group. Thus, section $M_1 \rightarrow 3 \rightarrow i_v \rightarrow 4$ is the external

constraint of the cutting motion group, section $4 \to 5 \to \sum_1 \to 6 \to i_x \to 9$ is part of *the internal constraint* of the cutting-speed motion group, and only section $9 \to i_s \to 10$ is an independent part of the drive gear train in the feed group. The use of certain chains of the *cutting speed motion* group as feed gear trains is an advantageous procedure since it simplifies the structural arrangement of a gear hobber with a single drive motor. However, if a considerable load is transmitted by the feed gear train, it may adversely affect the cutting speed motion group through the common branches of the constraints. As a rule, the feed group is not subject to a high load and its influence on the cutting speed group is negligible.

It has been established by means of structural analysis that there can be four possible versions of the structure of the formative part in gear hobbing *machines*. *Thus, they may have*

(1) two drive motors and no intergroup constraints (see Fig. 14f);
(2) two drive *motors and an intergroup constraint* (see Fig. 15a),
(3) one common drive motor and feed per minute (Fig. 15b),
(4) one common drive motor and feed per revolution (Fig. 15c).

Only the first and fourth structure versions of gear hobbers find application in practice.

Though the structure in each kinematic constraint may vary as to the number and type of members composing the constraint, the number of constraints, their type and their interconnection can be only as indicated in the four preceding versions if the gear hobber has a common movable operative member. Thus, *for example, cutting speed and feed gearboxes* have been installed in place of the cutting speed and feed change gears in the models 5K32 and 5K324 gear hobbers of the Komsomolets Plant (see Fig. 46), but the structure of the formative part still fully complies with the fourth structure version (Fig. 15c).

Next we shall consider the structure versions of the formative part of a machine tool which does not have any common movable operative members. To this end we shall take a machine tool *with the same number and type* of kinematic groups as those of the gear hobber considered previously with a structure of class C24, i.e. a machine tool *with a structure of a complex* class with two complex formative groups of four elementary motions and without a common movable *operative* member.

The kinematic interconnection of the groups, in this case, can be only along the external constraints, and there are four possible versions for the structure of the formative part of the machine. The structure may incorporate

(1) two drive motors and no intergroup constraint (Fig. 16a),
(2) two drive motors and an intergroup constraint (Fig. 16b),
(3) one common drive motor and feed per *minute* (Fig. 16c),
(4) one common drive motor and feed per revolution (Fig. 16d).

plex. This is due to the fact that the external and part of the internal constraint of the first kinematic group for the cutting-speed motion are used as the external constraint of the second, feed motion, group.

To establish the constitution of the feed group, we remove the cutting-speed motion group from the general structure of the formative part of the gear hobber (Fig. 15c). The remaining part of the structure refers to the feed motion group $F_s\,(T_3R_4)$. Representing this structure conventionally we obtain

$$\begin{tikzcd}
 & & & & R_4 & 9 \arrow[l] \arrow[r] & i_s \arrow[r] & 10 \arrow[r] \arrow[ddd] & 8 \arrow[r] & t_1 \\
 & & & & & 2 \arrow[u] & & & & \\
M^1 \arrow[r] & 3 \arrow[r] & i_v \arrow[r] & 4 \arrow[dd] & & & & & & \\
 & & & & & i_x \arrow[uu] & & i_y \arrow[dd, dash] & & \\
 & & & 5 \arrow[r] & \sum_1 \arrow[r] & 6 \arrow[u] & & & & \\
 & & & & 7 \arrow[u] & & & {} \arrow[lll] & &
\end{tikzcd}$$

This representation shows that motion is transmitted from drive motor M_1 through the external constraint of the feed group, speed change gears i_v, differential $\sum_1$, indexing change gears i_x, point *9*, and feed change gears i_s to point *10* which is the interconnection of the external and internal constraints in the feed group $F_s\,(T_3R_4)$. From point *10* motion is transmitted along the internal constraint of the feed group through point *8* and vertical feed screw to the hob slide which accomplishes the elementary motion T_3. Motion is likewise transmitted from point *10* through the differential change gears i_y, second input shaft *7* of the differential, differential $\sum_1$, output shaft *6* of the differential, indexing change gears i_x and point *2* to the work table to which the second elementary motion R_4 is imparted. This group produces the helical feed motion $F_s\,(T_3R_4)$. Since the feed change gears i_s are arranged following the differential and the indexing change gears i_x, the hobber has a feed per revolution which remains constant upon any change-overs of change gears i_x and i_y, if change gears i_s are not altered. The feed per minute depends upon setups of the change gears i_v, i_x and i_s.

This representative structure of the formative part (Fig. 15c) is applied in the most widely used gear hobbers in the Soviet Union (models 5Д32 and 5E32 of the Komsomolets Plant) as well as in many other Soviet and foreign models.

It is also evident from the structural diagram that the external constraint of the feed group, between motor M_1 and point *10*, is not an independent and separate feed gear train, but consists of a series of sections belonging to the cutting-speed motion group. Thus, section $M_1 \rightarrow 3 \rightarrow i_v \rightarrow 4$ is the external

constraint of the cutting motion group, section $4 \to 5 \to \sum_1 \to 6 \to i_x \to 9$ is part of the internal constraint of the cutting speed motion group, and only section $9 \to i_s \to 10$ is an independent part of the drive gear train in the feed group The use of certain chains of the cutting-speed motion group as feed gear trains is an advantageous procedure since it simplifies the structural arrangement of a gear hobber with a single drive motor However, if a considerable load is transmitted by the feed gear train, it may adversely affect the cutting speed motion group through the common branches of the constraints As a rule, the feed group is not subject to a high load and its influence on the cutting speed group is negligible

It has been established by means of structural analysis that there can be four possible versions of the structure of the formative part in gear hobbing machines Thus, they may have

(1) two drive motors and no intergroup constraints (see Fig 14f);
(2) two drive motors and an intergroup constraint (see Fig 15a),
(3) one common drive motor and feed per minute (Fig 15b),
(4) one common drive motor and feed per revolution (Fig 15c)

Only the first and fourth structure versions of gear hobbers find application in practice

Though the structure in each kinematic constraint may vary as to the number and type of members composing the constraint, the number of constraints, their type and their interconnection can be only as indicated in the four preceding versions if the gear hobber has a common movable operative member Thus, for example, cutting speed and feed gearboxes have been installed in place of the cutting speed and feed change gears in the models 5K32 and 5K324 gear hobbers of the Komsomolets Plant (see Fig 46), but the structure of the formative part still fully complies with the fourth structure version (Fig 15c)

Next we shall consider the structure versions of the formative part of a machine tool which does not have any common movable operative member To this end we shall take a machine tool with the same number and type of kinematic groups as those of the gear hobber considered previously with a structure of class C24, i e a machine tool with a structure of a complex class with two complex formative groups of four elementary motions and without a common movable operative member

The kinematic interconnection of the groups, in this case, can be only along the external constraints, and there are four possible versions for the structure of the formative part of the machine The structure may incorporate

(1) two drive motors and no intergroup constraint (Fig 16a),
(2) two drive motors and an intergroup constraint (Fig 16b),
(3) one common drive motor and feed per minute (Fig 16c),
(4) one common drive motor and feed per revolution (Fig 16d)

It is impossible to connect the two groups kinematically in the first version which has neither a common movable operative member nor a drive motor in common, since there are no common members in either the external or internal kinematic constraints. The two groups, in this case, can be interconnected only nominally, i.e. they can be mounted on a single member, such as the bed. The groups are interconnected kinematically in the second, third and fourth versions, and then only through external constraints. In actual practice, machine tools may be encountered with a common movable member as well as without one, therefore, the number of possible structure versions is limited to eight, of which one structure—with a common movable member and common drive motor and feed per minute (*Fig. 15b*)—is very rarely employed. Of the seven remaining structures, the most extensively applied, due to its high efficiency, is the one with a common movable member and a common drive motor employing feed per revolution (Fig. 15*c*).

From the comparison of the four structure versions of machine tools having a common operative member (*Fig.* 15) with those who do not (Fig. 16), it is evident that their difference lies primarily in that the groups are interconnected along internal constraints in the machine tools having a common operative member. Consequently, most of these machine tools have a summation mechanism (differential). In this case, there may or may not be interconnections along the external constraints, but interconnection along internal constraints exists without fail. If group interconnection along external constraints does occur, then the method of interconnection is the same in both types of machines. To get a better idea of the possible versions of kinematic structure of machine tools, it is necessary to consider the problem of arranging the setting-up devices.

The structural diagrams of gear hobbers (Fig. 17*a*) indicate that the internal and external gear trains in both groups consist of a series of sections. For instance, the internal constraint of the cutting motion group consists of five sections: *3-2*, *2-4*, *4-5*, *5-6* and *6-7*. In the given case, the indexing change gears i_x are arranged in section *5-6*, but could just as well have been arranged in any other section of the same chain. Sections *5-6* and *6-7* are common sections since they belong to the internal kinematic constraint of both the cutting and feed motion groups. Therefore, both the indexing change gears i_x and the differential change gears i_y can be arranged in these sections. The feed i_s and speed i_v change gears can also be otherwise arranged. Thus, by varying the arrangement of the four sets of change gears, a fairly large number of structure versions can be obtained. The most efficient arrangement of the setting-up devices is one enabling the machine to be changed over in cutting a pair of mating gears by the easiest procedure in the shortest period of time. An additional condition is that any changes made in i_v and i_s to select the most rational cutting speeds and feeds should entail no changes in i_x and i_y. Such an arrangement of the change gears is shown in *Fig. 17a*.

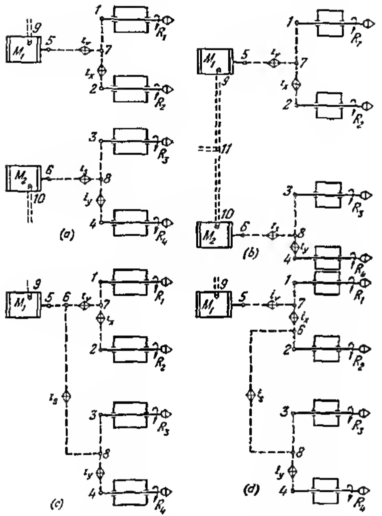
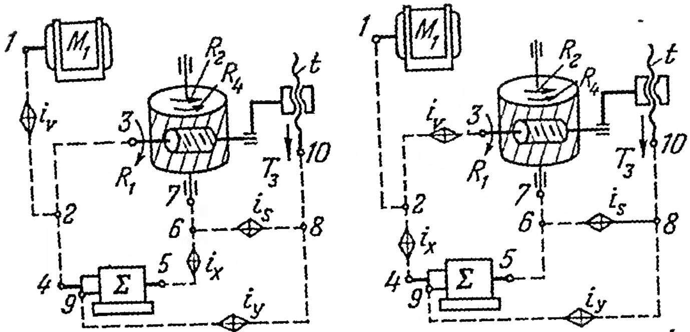


Fig 16 Structure versions of the formative part of a machine tool which does not have any common movable operative members in its system of motion

As a matter of fact, we shall try to move the change gears to other sections of the diagram If the cutting speed change gears i_v are arranged in section *3 2* (Fig 17*b*) it will be more difficult to set up the indexing change gears i_x since any changing-over of i_v will require a corresponding change-over in i_x.

If change gears i_x are arranged in the section *3-2* (Fig. 18*a*), then any changes made in them will require like changes in the speed change gears i_v



$$1 q_{3-2} q_{2-4} q_{sd} q_{5-6} i_x q_{6-7} = \frac{k}{z}$$

$$i_x = \frac{1}{q_{3-7}} \times \frac{k}{z}$$

$$1 q_{7-6} q_{6-5} \frac{1}{i_x} q_{sd} q_{9-8} \frac{1}{i_y} q_{8-10} t = \frac{\pi m z}{\sin\beta}$$

(a)

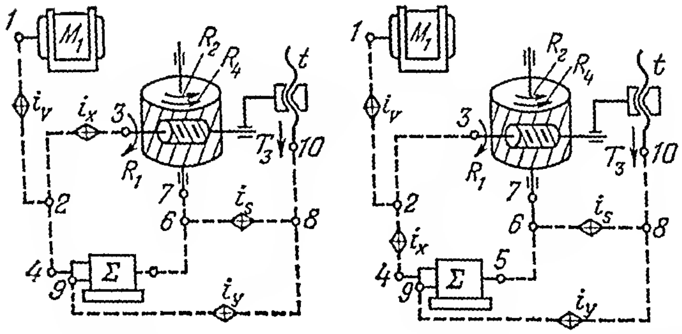
$$1 q_{3-2} \frac{1}{i_v} q_{2-4} q_{sd} i_x q_{5-6} q_{6-7} = \frac{k}{z}$$

$$i_x = \frac{i_v}{q_{3-7}} \times \frac{k}{z}$$

$$i_y = q' \frac{\sin\beta}{m}$$

(b)

Fig. 17. Versions of change gear arrangement



$$n_e q_{1-2} i_v q_{2-3} i_x = n_h$$

$$i_v = \frac{n_e}{q_{1-2} q_{2-3}} \times \frac{n_h}{i_x}$$

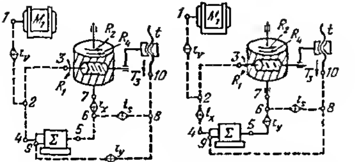
(a)

$$1 q_{7-6} q_{6-5} q_{sd} q_{9-8} \cdot \frac{1}{i_y} q_{8-10} \cdot t = \frac{\pi m z}{\sin\beta}$$

$$i_y = \frac{q_{7-10}\, t}{\pi} \times \frac{\sin\beta}{mz}$$

(b)

Fig. 18. Versions of change gear arrangement



$1 q_{7\,6} \frac{1}{i_x} i_s q_{6\,8} q_{8\,10} t = s$

$i_s = q'' \frac{sk}{z}$

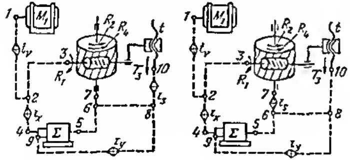
(a)

$1 q_{3\,2} q_{2\,4} i_x q_{5d} q_{5\,6} i_y q_{6\,7} = \frac{k}{z}$

$i_x = \frac{1}{q_{3\,7}} \frac{1}{i_y} \frac{k}{z} = q'' \frac{km}{\sin\beta}$

(b)

Fig 19 Versions of change gear arrangement



$i_y\ q' \frac{\sin\beta}{m} \times \frac{1}{i_s}$

(a)

$i_x = \frac{1}{q_{3\,7}} \times \frac{k}{z} \times \frac{1}{i_s}$

(b)

Fig 20 Versions of change gear arrangement

Moreover, change gears i_x, arranged within the cutting speed gear train, will be subject to higher loads. This cannot be permitted.

Let us assume that the indexing change gears i_x are in section *2-4* (Fig. 18*b*). Then, after cutting one gear of a pair and in changing over the machine to cut the mating gear, it will be necessary to change i_x (because the number of teeth z has changed) and i_y (since the lead P of the helical teeth has changed). This may result in an inaccurate lead P due to the approximate setting-up of change gears i_y and to loss of time in changing over the hobber to cut the second gear.

Another possibility is to arrange change gears i_x in section *5-6* (see Fig. 17*a*). In this case, the value z is cancelled in the set-up formula for i_y so that the ratio of these change gears will depend only upon the module m and helix angle β of the helical teeth, both of these values being identical for mating gears. Thus, only i_x need be changed over to cut the mating gear of a pair. Set-up time is reduced and a more accurate lead is obtained on the helical teeth.

The last possible location of the indexing change gears i_x is in the section *6-7* of the constraint (Fig. 19*a*). In this case, it will be necessary to change i_s each time the indexing change gears i_x are changed over. This also leads to a loss of time in changing over the hobber to cut a mating gear.

These two arguments exclude the feasibility of moving the differential change gears i_y to section *5-6* or *6-7* (Fig. 19*b*). The only remaining possibility to be checked is to move feed change gears i_s to section *8-10* (Fig. 20*a*). This proves inconvenient as i_y must be changed over if changes are made in i_s.

Hence, the most rational arrangement of the differential change gears is in section *8-9*.

The feed change gears i_s could be moved to section *6-7* (Fig. 20*b*). However, in selecting the most efficient feed and in changing over i_s, here being within the internal constraint, it would be necessary to change over two more sets of change gears, i_x and i_y. Such a procedure cannot be permitted under any circumstances.

It follows that the most efficient arrangement of the setting-up devices is the one shown in the structural diagram in Fig. 17*a*. This is why this arrangement is the one used in gear hobber design.

4-2. General Procedure for Analysing the Kinematic Structure of Machine Tools; Setting Up the Kinematic Chains

It is possible to analyse the kinematic scheme of any metal-cutting machine tool by applying the general theoretical principles concerning the kinematic structure of machine tools, formulated in the preceding section.

Such an analysis is based on the following three principles.

(a) The kinematic scheme is analysed in parts and not as a whole, the first to be considered being the kinematic groups producing the formative indexing and feed in motions and then the others such as the control and handling motion groups

(b) The analysis of the kinematic structure of each group is begun not with the sources of motion (motors) but with the kinematic chains or pairs which provide the internal kinematic constraint inside the kinematic group Only after this, the drive train from the motor is established

(c) Structural chains are considered in structural analysis, while in setting up the machine the kinematic calculation chains are dealt with Structural and calculation chains are not the same thing A structural kinematic chain is a real chain or gear train providing the kinematic constraints required to obtain the specified parameters for the operative motion being produced Kinematic calculation chains on the other hand, are drawn up with the sole aim of determining the unknown parameters of the setting up devices Consequently, calculation chains differ from structural chains both in constitution and in number

In the general case the analysis of the kinematic structure and the setting up of the kinematic chains in a machine tool can be broken down into four stages

1 Proceeding from the shape of the surface to be obtained the cutting tool to be used and process of shaping the material the number and type of kinematic groups are determined for the formative, indexing and feed in motions Then the kinematic scheme is divided into as many parts as there are groups

2 Knowing the nature of the operative motions the structure of each kinematic group is examined separately The internal and external con straints are established in each group and the devices are found for setting up and regulating the parameters of the operative motion

3 The remainder of the kinematic scheme usually consisting of the controls and the kinematic groups for handling motions is considered

4 The kinematic chains of the machine tool are set up after deriving the setup formulas for the change-gear sets and certain other setting up devices

The setup formula for any change gear set is determined by first outlining the kinematic calculation chain in accordance with the kinematic scheme of the machine The equation which is next worked out on the basis of the calculation chain has been called the "kinematic balance equation" by Prof G Golovin who first proposed a unified setup formula for all machine tools The ratio for each set of change gears is determined in this way Since several different kinematic calculation chains can sometimes be outlined for a single set of change gears these chains are only conventional ones which do not always determine the nature of the kinematic scheme in the

machine tool. Hence, kinematic calculation chains serve only to determine the ratios of the change gears. Machine tools of the same type with the same kinematic structure may have a different number of change-gear sets and different kinematic calculation chains if the parameters of the operative motion being produced are to be set up in one machine by means of change gears, and in the other by another type of setting-up device whose characteristics can be found without working out the kinematic balance equation.

To work out the kinematic balance equation of a calculation chain it is necessary to know the basic displacement of the final members. The procedure employed to determine these displacements depends upon whether the setting-up device being considered is in the internal or external kinematic constraint.

If the setting-up device is within the internal constraint, the final members will be movable operative members. The absolute displacements of the members are not known but their relative displacements are; they correspond to the relative displacements in a mechanical transmission whose form is a copy of the given workpiece and cutting tool.

To this end, a definite motion is imparted to one of the elements of the transmission (work or cutting tool): one revolution in rotation or a displacement of L mm in rectilinear motion.

On the basis of the transmission ratio between the work and tool (it is the same as in the transmission which in form is a copy of the work and cutting tool), the displacement of the second element (cutting tool or work) is determined.

For example, the basic displacement of the final members of the indexing chain in hobbing a spur or helical gear can be written as follows, taking into consideration that the gear blank and hob copy the motion of a worm drive, or transmission,

$$1 \text{ revolution of the hob} \longrightarrow \frac{k}{z} \text{ revolutions of the blank}$$

where k = number of starts on the hob
z = number of teeth on the gear being cut.

The calculation and structural chains of the external constraints usually differ in composition. The calculation chains for determining the ratio of the cutting-speed change gears are very simple in constitution. The kinematic balance equation links the motor speed (rpm) with the speed (rpm) of one of the operative members participating in the production of the cutting motion. The basic displacements in the cutting-speed chain of a gear hobber, for instance, are

$$n_e \text{ rpm of the motor} \longrightarrow n_h \text{ rpm of the hob}$$

The constitution of the calculation feed chains depends upon the unit of feed employed and this in turn depends upon the accepted manufacturing process. The different feed units are feed per revolution (mm per rev), feed per minute (mm per min) and cycle time (sec per cycle).

Thus, the basic displacements in the feed chains are:

1. For feed per minute

linear feed for motion of the $F_s(T_1)$ or $F_s(T_2R_3)$ type

n_e rpm of the electric motor $\rightarrow s$ mm per min of translatory motion of the operative member in the feed group

rotary feed for motion of the $F_s(R_1)$ or $F_s(R_2R_3)$ type

n_e rpm of the electric motor $\rightarrow n$ rpm of the movable operative member in the feed group

2. For feed per revolution

linear feed for motion of the $F_s(T_1)$ or $F_s(T_2R_3)$ type

1 revolution of the movable operative member in the cutting speed group	$\rightarrow s_r$	displacement of the movable operative member in the feed group
or		
1 full stroke (back and forth) of the movable operative member in the cutting speed group	$\rightarrow s_r$	displacement of the movable operative member in the feed group

rotary feed for motion of the $F_s(R_1)$ or $F_s(R_2R_3)$ type

1 revolution of the movable operative member in the cutting speed group	$\rightarrow \frac{s_r}{\pi D}$	revolutions of the movable operative member in the feed group
or		
1 full stroke (back and forth) of the movable operative member in the cutting speed group	$\rightarrow \frac{s}{\pi D}$	revolutions of the movable operative member in the feed group

3. For feed expressed as cycle time

1 revolution of the camshaft $\rightarrow \frac{n_e}{60} s_c$ revolutions of the motor

The kinematic structure of a machine tool depends not only upon the principles followed in linking the members but also on a great many other

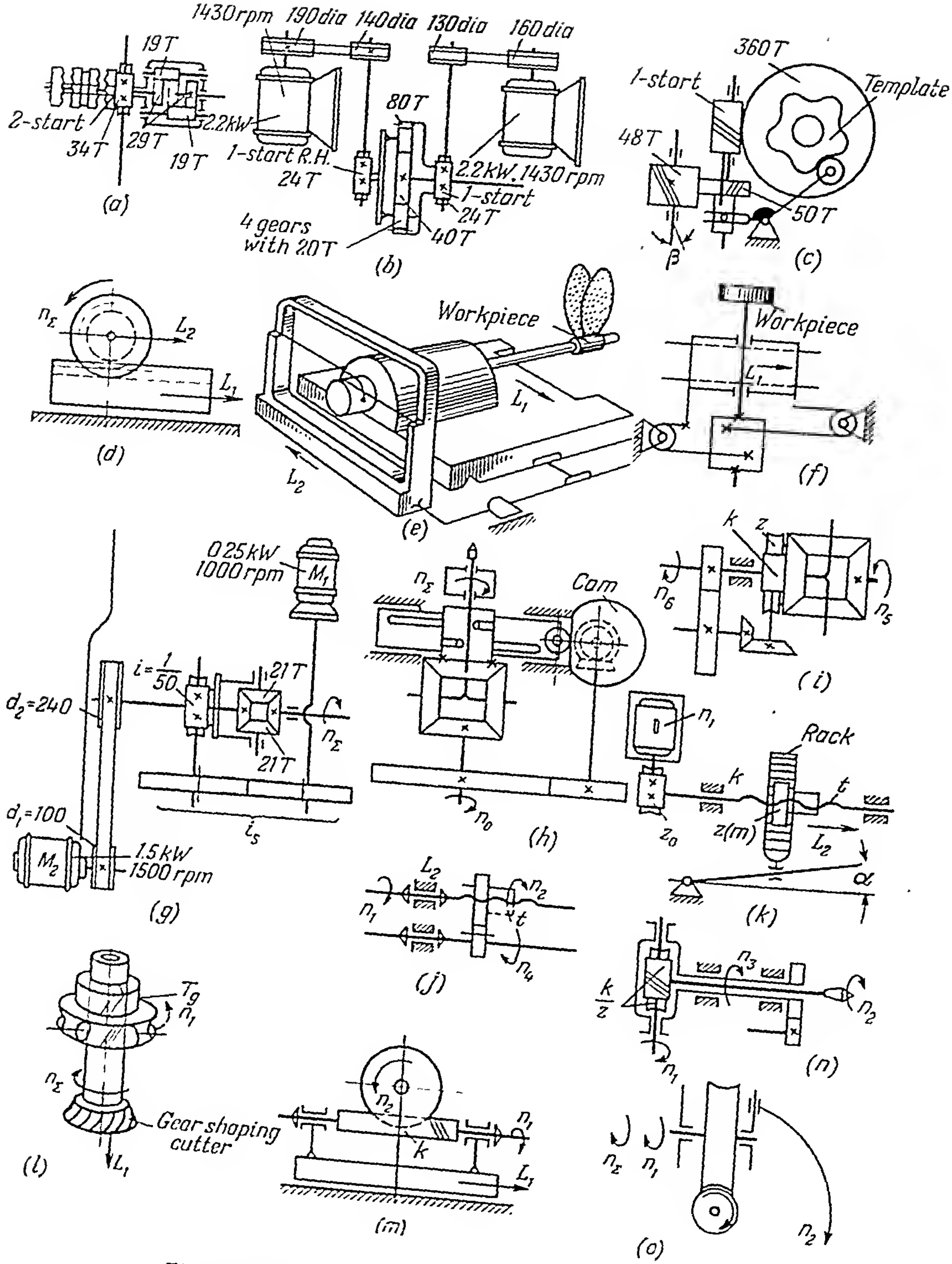


Fig. 21. Diagrams of summation mechanisms

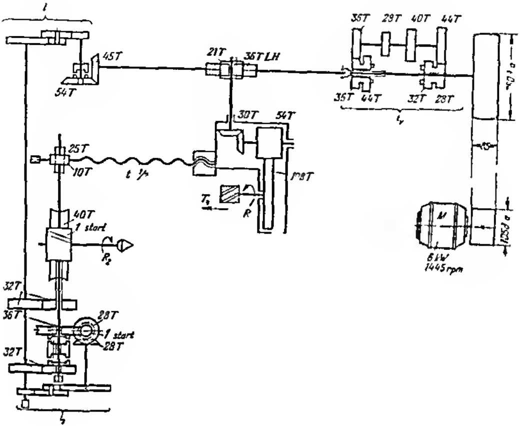


Fig 22 Kinematic diagram of the nondifferential gear hobber model 5A610

tial gear train while the feed gear train must be connected to the cutting speed group so that the internal constraint between the elementary motions making up the operative feed motion is accomplished through the cutting speed group

Shown in Fig 22 is the kinematic diagram of the horizontal type nondifferential gear hobbing machine model 5A610 This machine can conveniently hob spline shafts and wide spur or helical pinions integral with their shafts

The indexing change gears i_x are set up to cut a helical gear differently in this hobber than in a differential hobber

As a rule to set up the indexing change gears i_x one revolution of the gear blank is linked in the cutting speed motion group F_v $(R_1 R_2)$ with $\frac{k}{z}$

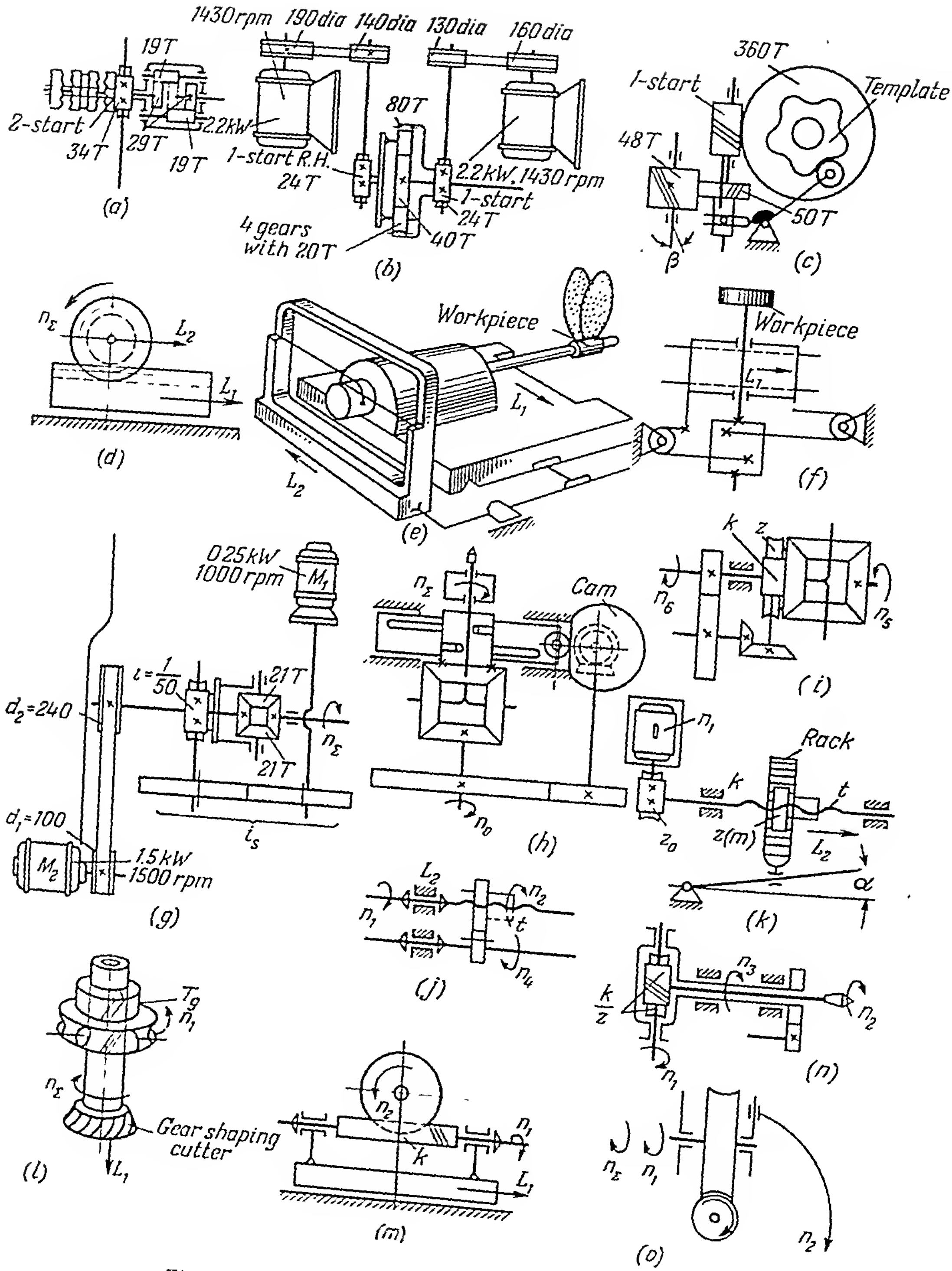


Fig. 21. Diagrams of summation mechanisms

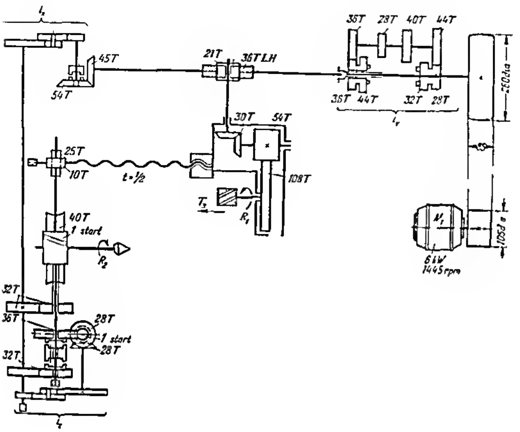


Fig 22 Kinematic diagram of the nondifferential gear hobber, model 5A610

tial gear train, while the feed gear train must be connected to the cutting speed group so that the internal constraint between the elementary motions making up the operative feed motion is accomplished through the cutting speed group

Shown in Fig 22 is the kinematic diagram of the horizontal type nondifferential gear-hobbing machine, model 5A610 This machine can conveniently hob spline shafts and wide spur or helical pinions, integral with their shafts

The indexing change gears i_x are set up to cut a helical gear differently in this hobber than in a differential hobber

As a rule, to set up the indexing change gears i_x, one revolution of the gear blank is linked in the cutting speed motion group F_v (R_1R_2) with $\frac{k}{z}$

revolutions of the hob, where z is the number of teeth on the gear being cut and k is the number of starts on the hob.

In nondifferential hobbing machines, rotation R_1 of the hob is linked with the summated rotation $R_2 \pm R_4$ of the gear blank, in which case the elementary motion R_4 belongs to the operative feed motion F_8 (T_3R_4) providing helical teeth with a lead of P mm. Therefore, hob rotation must be linked through the indexing train with the sum (or difference) of the two elementary rotary motions $R_2 \pm R_4$ of the gear blank.

Since the longitudinal feed s of the hob in a gear hobber is usually referred to one revolution of the gear blank, the hob must turn $\frac{z}{k}$ revolutions during one gear blank revolution to obtain the tooth profile, and $\pm\frac{s}{P}\times\frac{z}{k}$ revolutions to obtain the required tooth helix, the hob being fed in the longitudinal direction by an amount exactly equal to s. Consequently, the ratio i_x of the indexing change gears is determined on the basis of the following basic displacements of the final members in the indexing train:

$$1 \text{ revolution of the gear blank} \longrightarrow \frac{z}{k}\left(1 \pm \frac{s}{P}\right) \text{ revolutions of the hob}$$

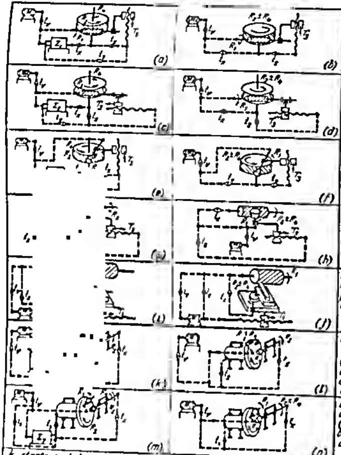
The sign to be used depends upon the combination of the hands of the helical gear and the hob thread.

Similar relationships can be derived for other types of machine tools as well. A number of differential gear-cutting machines are given in Fig. 23 that can be replaced by nondifferential machines.

It follows from the basic displacements given above that the shape of the surface being produced depends upon the rate of feed. Since change gears arranged within the internal constraints can only be set up with some approximation, due to complex setup formulas, it proves quite complicated to set up the kinematic chains of nondifferential hobbers. In some cases, this may lead to additional errors in the shape of the mating surfaces being machined, this being one of the essential drawbacks of machine tools with a nondifferential structure. On the other hand, these machines can operate with high kinematic accuracy, due to their higher torsional rigidity.

The indicated operational advantages and disadvantages of nondifferential machine tools determine their limits of application. They are used in large-lot and mass production where machining conditions are maintained constant and where the use of special change gears enables mating surfaces of the required shape to be obtained.

Machine tools with a differential structure can easily be changed over from job to job, and are therefore suitable for piece and small-lot production as well.



Operation performed	Operative motions	Differential machine tool	Nondifferential machine tool
Cutting helical gear with a gear hob	$F_v(R_1R_2)$ $F_s(T_3R_4)$	1 rev of blank $\rightarrow \frac{z_{gb}}{k}$ rev of hob 1 rev of blank $\rightarrow$ P mm longitudinal travel of hob	1 rev of blank $\rightarrow \frac{z}{k}(1\pm\frac{s_{lg}}{P})$ rev of hob $\rightarrow s_l$ mm longitudinal travel of hob
Cutting worm wheel with a gear hob by tangential feed	$F_v(R_1R_2)$ $F_s(T_3R_4)$	1 rev of blank $\rightarrow \frac{z_{gb}}{k}$ rev of hob L mm axial travel of hob $\rightarrow \frac{L}{\pi m z_{gb}}$ rev of blank	1 rev of blank $\rightarrow \frac{z_{gb}}{k}(1\pm\frac{s_a}{\pi m z_{gb}})$ rev of hob $\rightarrow s_a$ axial travel of hob
Generating a helical gear with a rotary tool (gear shaping cutter)	$F_v(R_1R_2)$ $F_s(R_3R_4)$	1 rev of blank $\rightarrow \frac{z_{gb}}{z_{sc}}$ rev of cutter 1 rev of blank $\rightarrow$ P longitudinal travel of cutter	1 rev of blank $\rightarrow \frac{z_{gb}}{z_{sc}}(1\pm\frac{s_{lg}}{P})$ rev of cutter $\rightarrow s_{lg}$ mm longitudinal travel of cutter
Generating a multiple start cylindrical worm with a rotary tool (gear shaping cutter)	$F_v(R_1R_2)$ $F_s(T_3R_4)$	1 rev of blank $\rightarrow \frac{z_{gb}}{z_{sc}}$ rev of cutter 1 rev of blank $\rightarrow$ P mm longitudinal travel of cutter	1 rev of blank $\rightarrow \frac{z_{gb}}{z_{sc}}(1\pm\frac{s_{lg}}{P})$ rev of cutter $\rightarrow s_{lg}$ mm longitudinal travel of cutter
Relieving a helical flute on a milling cutter with a single point tool	$F_v(R_1R_2)$ $F_s(T_3R_4)$	1 rev of blank $\rightarrow \frac{z_{cb}}{p}$ rev of cam 1 rev of blank $\rightarrow$ P mm longitudinal travel of blank	1 rev of blank $\rightarrow \frac{z_{gb}}{p}(1\pm\frac{s_{lg}}{P})$ rev of cam $\rightarrow s_{lg}$ mm longitudinal travel of cutter
Cutting a curved tooth (epicycloid) bevel gear with a face-mill cutter	$F_v(R_1R_2)$ $F_s(R_3R_4)$	1 rev of cutter $\rightarrow \frac{z_t}{z_{gb}}$ rev of blank 1 rev of cutter $\rightarrow \frac{z_t}{z_{cg}}$ rev of cradle	1 rev of cutter $\rightarrow \frac{z_t}{z_{gb}} \pm s_{cd}\frac{z_{cg}}{z_{gb}}$ rev of blank $\rightarrow s_{cd}$ rev of cradle
Cutting a curved tooth (involute) bevel gear with a conical hob	$F_v(R_1R_2)$ $F_s(R_3R_4)$	1 rev of hob $\rightarrow \frac{k}{z}$ rev of blank 1 rev of cradle $\rightarrow \frac{z_{cg}}{z_{gb}}$ rev of blank	1 rev of hob $\rightarrow \frac{k}{z_{gb}} \pm s_{cd}\frac{z_{cg}}{z_{gb}}$ rev of blank $\rightarrow s_{cd}$ rev of cradle
		Interdependence of elementary motions in the internal constraints	

k starts on hob
p number of lobes on cam
z_{gb} number of teeth to be cut
z_{sc} number of teeth on gear shaping cutter
z_t number of teeth skipped in indexing cycle
z_{cg} number of teeth on crown gear
P lead of the helix
s_{lg} longitudinal feed
s_a axial feed, mm/rev
s_{cd} cradle feed, rev per tool rev

Fig 23 Differential and nondifferential kinematic structures of machine tools

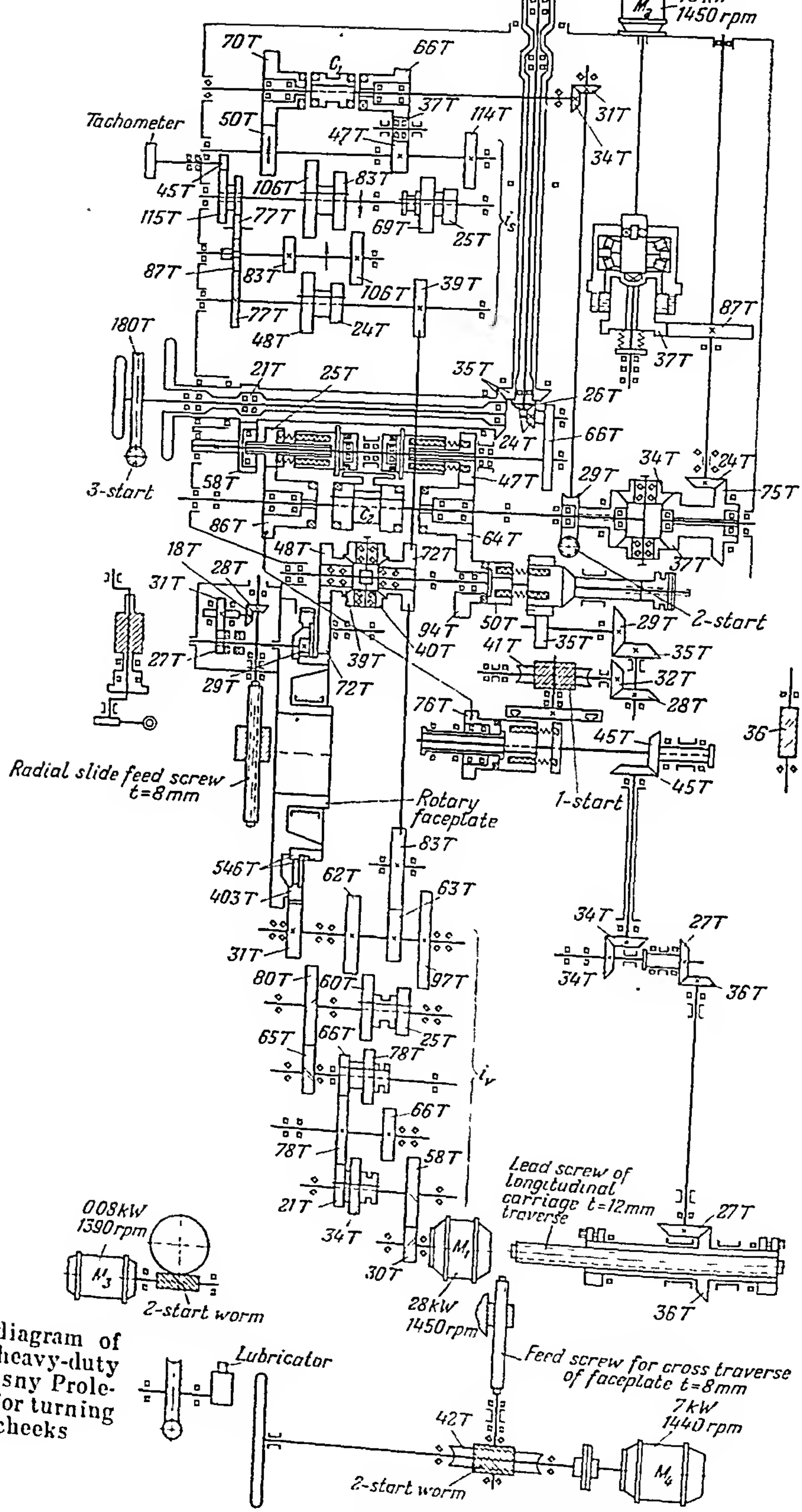


Fig. 25. Kinematic diagram of the model MK138M heavy-duty crankpin lathe (Krasny Proletary Plant, Moscow) for turning crankpins and cheeks

A motion of the same speed but in the opposite direction is imparted to the radial slide when pinion 29T rolls around gear 546T. Thus the radial slide receives the specified slow feed motion from the feed gearbox through the left hand differential (through its carrying arm). Hence the installation of the second (left hand) differential is due wholly to the accepted arrangement of the lathe in which rotating radial slides are employed. This leads to the automatic development of a hidden summation transmission in the drive of the radial slides.

Even though it has two differentials the kinematics of the lathe in this arrangement is simpler for machining heavy crankshafts than it would be with other arrangements in which the crankshaft rotated. The right hand differential enabling rapid traverse motions to be engaged at any moment in operation could be replaced by an overrunning clutch but the latter is less suitable in turning heavy crankshafts due to its poorer performance under severe dynamic conditions.

The radial working feeds are determined from the kinematic balance equation

$$1 \text{ revolution of the faceplate} \times \frac{403}{34} \times \frac{63}{33} \times i_s \times \frac{50}{0} \times \frac{34}{31} \times$$

$$\times \frac{2}{33} \times \frac{1}{2} \times \frac{6}{34} \times \frac{1}{-} \times \frac{78}{46} \times \frac{56}{23} \times \frac{27}{34} \times \frac{18}{28} \times 8 = s_{rd}$$

Longitudinal displacements of the carriage are calculated from the formula

$$1 \text{ revolution of the faceplate} \times \frac{403}{34} \times \frac{63}{39} \times i_s \times \frac{50}{0} \times \frac{34}{31} \times$$

$$\times \frac{2}{39} \times \frac{1}{2} \times \frac{64}{47} \times \frac{25}{86} \times \frac{86}{6} \times \frac{4}{5} \times \frac{34}{34} \times \frac{27}{36} \times \frac{27}{36} \times 12 = s_{lg}$$

If the change in one of the motion parameters due to presence of a hidden summation transmission is only slight and the motion produced is of a uniform nature there will be no need for eliminating the supplementary motion from the hidden transmission by means of a special differential complicating the kinematic structure of the machine. In this case it is better to take the supplementary motion into account in setting i
in the internal constraint of the feed group
in the kinematics of the spiral bevel gear g
considered

Spiral Bevel Gear Generator

The generator (Fig. 26) is intended for cutting curved tooth bevel gears up to 110 mm in diameter of a module from 0.5 to 1.5 mm and having from 5 to 80 teeth. The teeth are cut with two face mill type cutters each carrying

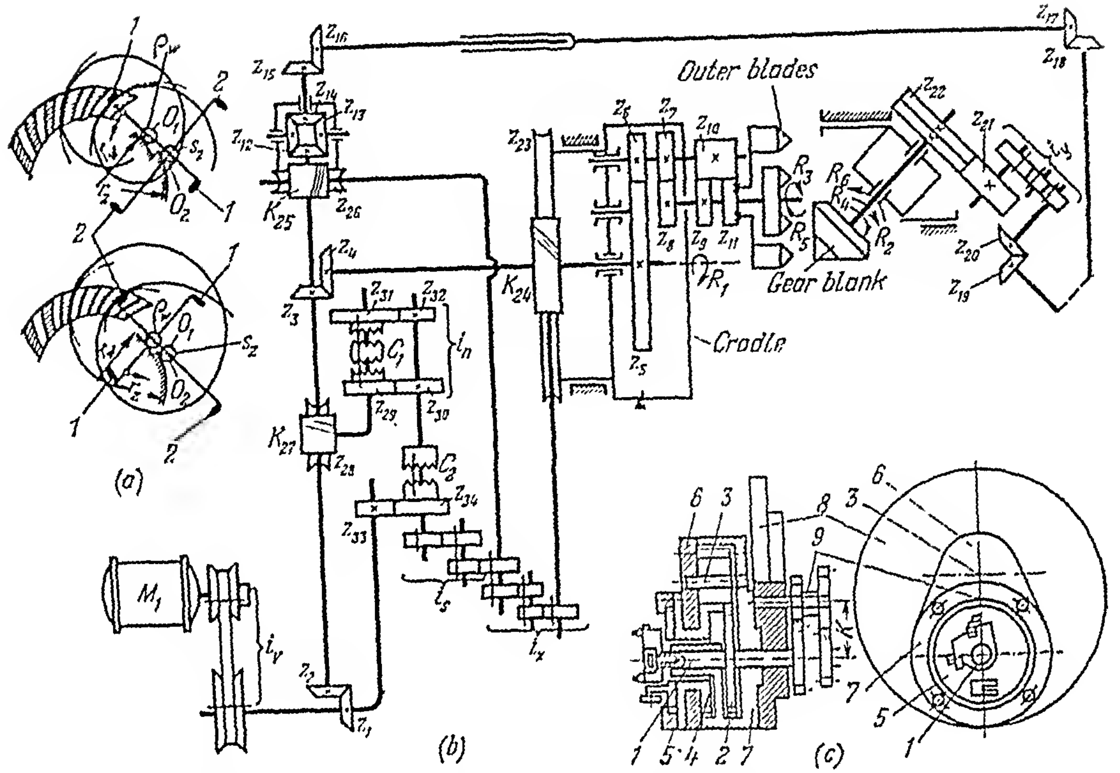


Fig. 26. Kinematic diagram of the spiral bevel gear generator, model K41A (Klingelnberg Co., FRG) for cutting curved-tooth bevel gears

two circular tools (blades) with straight cutting edges. An auxiliary blade for machining the bottom land of the tooth spaces is mounted on the inner cutter. The tooth profile is produced by the generating method as in all bevel gear cutting machines in which the gear blank, in its relative motion, rolls without slipping around a crown generating gear. The straight cutting edge of the blades represents one tooth of the imaginary crown gear. The generating motion $F_{к}(R_1 R_2)$, which is the slow feed motion, is produced by two co-ordinated elementary motions—cradle rotation R_1 and gear blank rotation R_2 (Fig. 26*b*).

The lengthwise form of the teeth depends upon the lengthwise form of the crown gear teeth which is a prolate epicycloid in this generator. Fig. 26*a* shows the curves for generating the convex and concave sides of the gear teeth.

Upon roll of a circle of radius ρ_w, belonging to the cutter, around a circle of radius r_w, blade *1* describes a prolate epicycloid on the crown gear (shown

at the top). The same is shown at the bottom for the concave side of the teeth which is cut with the outer cutter.

The roll motion of the cutters on the crown gear is the cutting motion $F_v\,(R_3R_4)$ which is accomplished in this generator by co-ordinated rotation R_3 of the cutter and rotation R_4 of the gear blank (Fig. 26b) Hence, two formative motions $F_v\,(R_3R_4)$ and $F_s\,(R_1R_2)$ are required to cut the gear teeth These motions are produced by two complex kinematic groups. The first of these, the cutting motion group $F_v\,(R_3R_4)$ consists of internal and external kinematic constraints

The internal constraint (Fig. 26b) interconnects the cutter and gear blank through spur gearing $\frac{z_{11}}{z_9}\times\frac{z_8}{z_7}\times\frac{z_6}{z_5}\times\frac{z_4}{z_3}$, a differential, bevel gearing $\frac{z_{15}}{z_{16}}\times\frac{z_{17}}{z_{18}}\times\frac{z_{19}}{z_{20}}$, indexing change gears i_y and the gears $\frac{z_{21}}{z_{22}}$, providing blade displacement along the specified path (a prolate epicycloid)

Through the external constraint $\left(\text{infinitely variable drive } i_v \text{ and gear } \frac{z_1}{z_2}\right)$ motion is transmitted from motor M_1 to the internal constraint The variable speed V-belt drive i_v with two adjustable pulleys provides a stepless cutter speed range from 100 to 400 rpm

The second kinematic group—the feed motion group—produces the generating motion $F_s\,(R_1R_2)$ Its internal constraint connects the cradle to the gear blank through worm gearing $\frac{z_{23}}{k_{24}}$, profiling change gears i_x, worm gearing $\frac{k_{25}}{z_{26}}$, differential bevel gearing, indexing change gears i_y and gears $\frac{z_{21}}{z_{22}}$

Motion is transmitted from motor M_1 to the internal constraint through variable speed drive i_v, bevel gearing $\frac{z_1}{z_2}$ worm gearing $\frac{k_{27}}{z_{28}}$, clutch C_1 and further through gears $\frac{z_{31}}{z_{32}}$ or $\frac{z_{29}}{z_{30}}$ and feed change gears i_s

The driven gear of these change gears is the member that connects the external and internal constraints Rapid rotation of the cradle can be obtained through gears $\frac{z_{33}}{z_{34}}$ and clutch C_2 It is used to set the cradle into the initial position The preceding constitutes the whole of the formative part of the generator The feed in group and other devices concerned with machine control are not shown in the diagram

In addition to the usual differential with bevel gears z_{12}, z_{13} and z_{14}, the kinematic scheme also incorporates a hidden differential mechanism Upon roll of the cradle, gear z_6, mounted in the cradle rolls along gear z_5 whose bearings are mounted in the base this leads to supplementary rotation R_5 of the cutters To obtain a tooth of proper shape, this supplementary rotation must be taken into consideration in setting up the generator, and a motion

R_6 must be transmitted to the gear blank. This motion R_6 must be co-ordinated with motion R_5 in the same proportion existing between the motions R_3 and R_4.

Motion F_v (R_3R_4) will also accomplish the indexing process. Depending upon the radii ϱ_w and ϱ_z of the rolling circles, the cutter will either cut tooth after tooth, consecutively, or it will skip several (z_i) teeth. The cutter shown in Fig. 26*c* skips two teeth each time ($z_i = 2$). In this generator z_i may vary from one to three teeth. Two sizes of cutters are used, the radius of the circle on which the inside blades are located being either 25 or 40 mm.

To ensure proper mating of the concave and convex sides of the teeth on two gears of a pair and to obtain suitable tooth bearing, it is necessary that the inside and outside cutter blades rotate about different centres, O_1 and O_2 (Fig. 26*a*). Because of this, two cutters are employed in the given machine, and the cradle consists of several rotary components (Fig. 26*c*). Plate *6* can be turned about pivot *3* to shift the centre of rotation of the outside blades in reference to that of the inside blades. To set the cutters to the radius vector $r = r_w + \varrho_w$ in reference to the cradle centre, eccentric member *8* is turned about pin *9*. It is also possible to turn housing *7* about shaft *1*. These settings enable teeth of the required shape, with correct tooth bearing, to be cut.

The procedure for setting up the kinematic chains of the spiral bevel gear grinder (Fig. 26*b*), taking into consideration the hidden summation transmission, follows.

I. *Setting up the cutting motion kinematic group* F_v (R_3R_4).

(a) Indexing change gears i_y are used to set up the required path of motion F_v.

First we determine the basic displacement of the final members of the kinematic calculation chain for indexing, and the kinematic balance.

The basic displacements are

$$1 \text{ revolution of the cutter} \rightarrow \frac{z_i}{z_{gb}} \text{ revolutions of the gear blank}$$

where z_i = number of teeth skipped in one revolution of the cutter

z_{gb} = number of teeth to be cut.

The kinematic balance equation is

$$1 \times \frac{z_{11}}{z_9} \times \frac{z_8}{z_7} \times \frac{z_6}{z_5} \times \frac{z_4}{z_3} \times \frac{z_{12}}{z_{14}} \times \frac{z_{15}}{z_{16}} \times \frac{z_{17}}{z_{18}} \times \frac{z_{19}}{z_{20}} \times i_y \times \frac{z_{21}}{z_{22}} = \frac{z_i}{z_{gb}}$$

Hence

$$i_y = c_1 \frac{z_i}{z_{gb}} \qquad (2)$$

where c_1 is the ratio of the constant components in the indexing train

$$c_1 = \frac{z_9 z_7 z_5 z_3 z_{14} z_{16} z_{18} z_{20} z_{22}}{z_{11} z_8 z_6 z_4 z_{12} z_{15} z_{17} z_{19} z_{21}}$$

The values z_i and z_{gb} must not have any common factors as otherwise it will not be possible to cut all the teeth on the gear by the generator

(b) The cutting speed motion F_v is set up by means of the variable-speed drive i_v We determine i_v

The kinematic balance equation is

$$n_e i_v \times \frac{z_1}{z_2} \times \frac{z_3}{z_4} \times \frac{z_5}{z_6} \times \frac{z_7}{z_8} \times \frac{z_9}{z_{11}} = n_c$$

and

$$i_v = c_2 n_c$$

where n_e = speed of the electric motor, rpm

n_c = speed of the cutter, rpm

c_2 = ratio of constant components in the cutting speed train

II *Setting up the feed motion kinematic group F_s ($R_1 R_2$)*

(a) Change gears i_x are used to set up the required path of motion F_s The basic displacements of the final members, in the kinematic calculation chain for profiling the teeth, are equal to

$$1 \text{ revolution of the cradle} \rightarrow \frac{z_{cg}}{z_{gb}} \text{ revolutions of the gear blank}$$

A correction must be introduced in these displacement calculations to compensate for the hidden differential drive *During* one revolution of the cradle, in the roll of gear z_6 about z_5 the cutter makes $1 \times \frac{z_5}{z_6} \times \frac{z_7}{z_8} \times \frac{z_9}{z_{11}} = c_3$ revolutions in reference to the cradle, while for c_3 revolutions of the cutter the gear blank must turn $c_3 \frac{z_i}{z_{gb}}$ revolutions

Therefore

$$1 \text{ revolution of the cradle} \rightarrow \frac{z_{cg}}{z_{gb}} \pm c_3 \frac{z_1}{z_{gb}} \text{ revolutions of the gear blank}$$

The kinematic balance equation is

$$1 \times \frac{z_{23}}{k_{24}} \times \frac{1}{i_x} \times \frac{k_{25}}{z_{26}} \times 2 \times \frac{z_{15}}{z_{16}} \times \frac{z_{17}}{z_{18}} \times \frac{z_{19}}{z_{20}} \times i_y \times \frac{z_{21}}{z_{22}} = \frac{z_{cg}}{z_{gb}} \pm c_3 \frac{z_i}{z_{gb}}$$

Hence

$$i_x = \frac{c_4 z_1}{z_{cg} \pm c_3 z_i} \tag{3}$$

where c_3 and c_4 = constant factors

(b) Setting up the rate of feed motion F_s

Upon rotation of the cradle through the angle $\theta°$, coinciding with the moments that the blades enter and leave the cutting zone, the teeth will

be completely cut in t seconds. The generator is set up to a specific cycle time by means of countergear i_{ct} and the feed change gears i_s.

The basic displacements are

$$\frac{n_e}{60} t \text{ rev of the electric motor} \rightarrow \frac{\theta}{360} \text{ rev of the cradle}$$

The kinematic balance equation is

$$\frac{n_e}{60} \times ti_v \times \frac{z_1}{z_2} \times \frac{k_{27}}{z_{28}} \times i_{ct} \times i_s \times i_x \times \frac{k_{24}}{z_{23}} = \frac{\theta}{360}$$

This equation is used to find i_s.

These spiral bevel gear generators are employed in instrument manufacture. Their advantages over straight and other spiral bevel gear generators is in the continuous nature of the cutting process and in the small number of simple circular tools (blades) employed in the cutters. These machines find efficient application in both piece and mass bevel gear production.

In certain gear generators of this type, the cradle operates with non-uniform rotation, and the elimination of motion from a hidden summation device by means of an appropriate kinematic setup is not feasible. In this case, a supplementary differential is built into the machine as has been done in the model 5284 * gear generator for cutting curved-tooth bevel gears.

4-5. Effect of the Shape and Size of the Cutting Tool on the Kinematic Structure of a Machine Tool

The shape of the cutting tool also has a definite influence on the kinematic structure of a machine tool. Insofar as kinematics is concerned, of primary interest in a cutting tool are those parameters that affect the number and type of operative motions of a machine tool and, consequently, its kinematic structure. Such parameters include the configuration of the cutting edge, its functions (purpose), the number of cutting edges and their relative arrangement on the cutting tool. The other parameters, such as the cutting angles, material, construction, sharpening procedure, etc., although they are of great importance in themselves, are concerned more with the cutting process as such and are not therefore to be treated in this section.

The configuration of the cutting edge (cutting profile) and its influence on the number and type of operative motions were considered previously

* The structural analysis and setting-up procedure for this model are considered in detail in *Kinematic Constraints in Metal-Cutting Machine Tools* by A. Fedotyonok, Mashgiz, Russian ed., 1960, pp. 241-247.

(see Fig. 1). It was established that, in this sense, there can be three possible types of cutting edges the cutting profile is a material line coinciding, in *one case, and not coinciding, in the second*, with the configuration and extent of the line being produced, or the profile may be a pointed edge in the form of a material cutting point In the first of these cases, the corresponding machine tool requires the minimum number of formative motions—one or two—while, in the other two cases, more formative motions—two or three—are required Hence, the kinematic structure of a machine tool based on the first case is always simpler than one based on the second or third case since it has less formative kinematic groups It should be emphasized, however, that the cutting tool required in the first case is more complex, as a rule, than that required in the third and even in the second case

The provision of groups of cutting edges on the cutting tool, each having its own, *separate purpose, such as shaping, indexing and feed in*, enables the available formative motions to be utilized for these processes instead of providing special operative motions As a result, the number of kinematic groups is reduced and the kinematic structure of the machine tool is simpler

The number and relative arrangement of the cutting profiles on the tool have an influence on the kinematic structure of a machine tool A single profile tool usually requires a greater number of motions to produce a certain surface than a multiple-profile tool

Examples in which the shape and size of the cutting tool affect the operative motion and, consequently, the kinematic structure of a machine tool, are given in Figs 27, 28 and 29

Figure 27 illustrates motion diagrams in milling threads with an ordinary thread-milling cutter having annular threads (Fig 27*a*) and with one having helical threads (Fig 27*b*) the helicoid thread surface being produced in the latter case by the forming and tangent method Theoretically, this method requires two formative motions—cutter rotation F_c and helical motion of the work F_s, as in milling thread with a cutter having annular threads (Fig 27*a*) In the cutter with annular threads the formed cutting edges, of a shape corresponding to the thread groove are arranged annularly around the circumference of the cutter Therefore upon cutter rotation, the generating profile has no supplementary axial displacement In a cutter with helical thread (Fig 27*b*), the cutting edges are arranged on a helix having *a pitch equal to that of the thread* to be milled Rotation of a cutter with this cutting edge arrangement leads to axial displacement of the generating profile, so that cutter rotation merges with the helical feed motion In this way, a single cutting speed motion F_v remains It is composed of interrelated rotation of the *cutter and* work at the same angular velocity $n_{gb} = n_c$ Here cutter rotation is a combined motion and therefore one of the two formative motions can be eliminated

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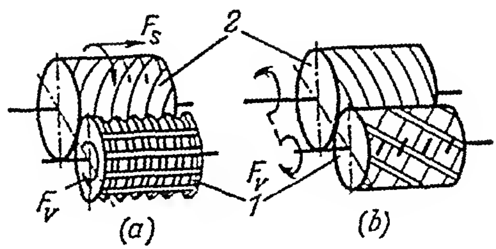

Fig. 27. Examples in which the shape and size of the cutting tool affect the operative motions in thread milling

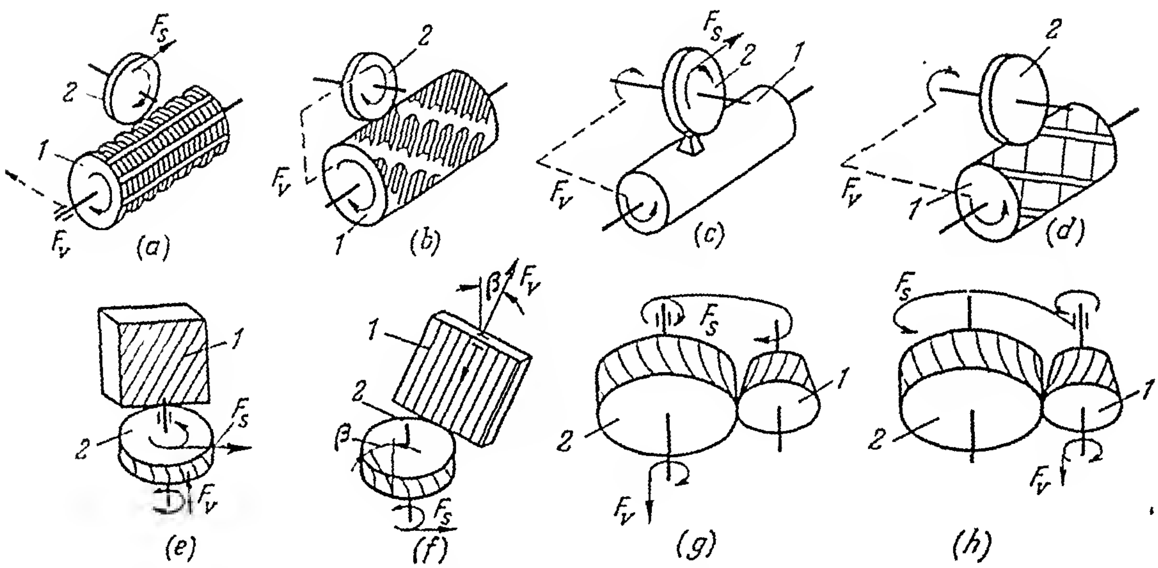

Fig. 28. Examples in which the shape and size of the cutting tool affect the operative motions in gear cutting

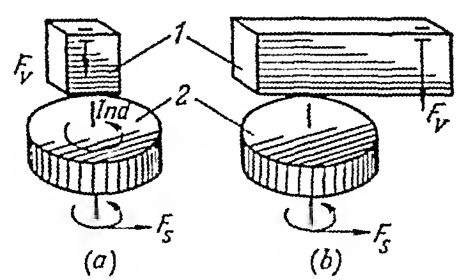

Fig. 29. Examples in which the size of the cutting tool affects the operative motions

Narrow face spur gears can be generated with a cylindrical rack type milling cutter (Fig 28a) having an annular thread under the condition that two formative motions are provided cutter rotation F_r and rolling of the gear blank along the cutter F_s. The side surfaces of the gear tooth are obtained by the tracing and generating method and not by the tangent and generating method since there is no tangential motion of the cutter (the motion shown by a dash line and arrow in Fig 28a) The tooth depth is not uniform over the face width of the gear the bottom of the tooth space is slightly concave along a circular arc and not straight as it would be if a tangential motion were provided If under the same conditions the same gear is cut with an ordinary gear hob having a helical thread (Fig 28b) hob rotation F_v merges with the rolling motion F_s of the blank and only one formative motion F_v remains (Fig 28b) This motion is composed of interrelated rotation of the hob and blank Thus a single formative motion F_v is substituted for the two motions F_v and F_s

The same occurs when a worm wheel is cut with a single tooth fly cutting hob (fly cutter) or with a gear hob (Fig 28c and d) In the latter case (Fig 28d) a single motion F_v composed of two interrelated rotary motions of the hob and gear blank shapes the tooth along the face width and tooth profile providing simultaneously for continuous indexing The gear teeth are machined by the tracing and generating method On the other hand two motions are required to cut the worm wheel with a fly cutter whose cutting edges represent one tooth of the hob (Fig 28c) The shape of the tooth along the face width of the worm wheel is produced and the indexing process is accomplished by motion F_v (rotation of the cutter coordinated with rotation of the blank) while the rolling motion F_s (rolling motion of the blank along the fly cutter) shapes the tooth profile

There are no combined formative motions in this case and therefore the actual number of formative motions coincides with the theoretical number

Cutting a helical gear with a rack type shaping cutter (Fig 28e and f) is a somewhat different matter In cutting the teeth by the tracing and generating method (Fig 28e) two motions are required the helical motion of the gear blank F_v which produces the helix along the face width of the gear and the rolling motion F_s of the blank along the cutter to obtain the tooth profile In this case each motion produces one geometrical generating line

If however the cutter travels in a direction inclined to the work axis (at the angle β Fig 28f) the tooth helix will be produced by the tangent method in which the required helix is tangent to a series of straight lines produced in turn by the reciprocating motion F_v of the rack type cutter This method nevertheless requires two motions the second being the rolling motion F_s of the blank along the cutter This second motion also gener

ates the tooth profile. Thus, it is the combined formative motion. Hence, the tangent and generating method theoretically requires three formative motions but, due to a combination of the shaping processes in which both geometrical generating lines are produced by a single motion, only two formative motions are actually necessary.

The same type of combined formative process is observed in cutting a helical gear with a rotary gear-shaping cutter (Fig. 28*g* and *h*). The same motions are shown in both of these diagrams: helical motion F_v and the rolling motion F_s of the blank about the cutter. The two diagrams differ in that the helical motion F_v in Fig. 28*g* is accomplished by the gear blank, and the side surfaces of the teeth are produced by the tracing and generating method. In Fig. 28*h*, this helical motion is accomplished by the cutter, and the helix along the tooth length is produced by the tangent and not the tracing method. The tangent method requires two formative motions F_v and F_s, and the tooth helix is tangent to the geometrical helixes produced by the helical motion of the cutter. Motion F_s is a combined motion as it also produces the tooth profile.

A single motion can accomplish and combine not only formative processes, but indexing processes as well. Thus, in cutting a spur gear with a short rack-type shaping cutter (Fig. 29*a*), in addition to the two formative motions, an indexing motion *Ind* (blank rotation) is also required. If a cutter of a length equal to or longer than the perimeter of the gear blank is used (Fig. 29*b*), then the indexing motion *Ind* becomes unnecessary. The rolling motion F_s provides for both the profiling of the teeth and indexing.

It follows from the preceding that the kinematic structure of a machine tool can be more simple or more complicated depending upon the shape and size of the cutting tool, since this may dictate the number and types of operative motions that are required. The simplest kinematic structure, however, is not the one to be preferred in all cases. The relationship between the cutting tool and the machine tool is such that the simpler the kinematic structure of the machine tool, the more complex and expensive is the cutting tool, not to mention the influence of other vital factors such as the required accuracy, output, manufacturing costs, etc. Hence, the engineering and economic aspects of this question must be carefully analysed in each particular case. The succeeding chapters of this part deal with the concrete application of the methods given above for analysing the kinematic structures and for setting up the kinematic chains of existing machine-tools.

The general method of kinematic structure analysis for machine tools proposed here enables the essence and specific features of the kinematics in a machine tool to be disclosed, without regard for their complexity, on the basis of its kinematic diagram.

CHAPTER 5

ANALYSIS OF THE KINEMATIC SCHEMES OF MACHINE TOOLS HAVING MECHANICAL KINEMATIC CONSTRAINTS

This chapter deals with the analysis of machine tools whose kinematic schemes most clearly illustrate the theoretical propositions concerning the kinematic structure of metal cutting machine tools

Machine tools with elementary kinematic structures, classes E11, E22 and E33, consisting of only simple kinematic groups that produce simple (rectilinear or rotary) operative motions in shaping surfaces, are not to be treated here since Part One (Vol 1) is devoted to general purpose machine tools, consisting of simple formative kinematic groups

5-1. Kinematic Structure of Thread-Making Machines

Methods of Making Threads

The kinematic structure of thread-making machines must consist of kinematic groups providing operative motions that produce a helix If the surface being produced is the trace obtained in the relative motion of the cutting edge, only a single complex helical motion is required Thus, thread is cut by a single point threading tool (Fig 30a), threading die or tap (Fig 30b) with a single helical operative motion F_r (R_1T_2)

The kinematic structure of machine tools that cut threads with a single point threading tool or a tap (Fig 31a and b) consists of a single kinematic group producing a helical motion F_r (R_1T_2) of the cutting edge on the tool in relation to the work (or vice versa) According to the diagram in Fig 31a, the internal kinematic constraint between the work spindle and the carriage comprises the threading gear train with setting up device i_x, and the drive gear train with setting up device i_v to obtain the required cutting speed

The threading gear train is usually absent in a tapping machine (Fig 31b), the internal constraint for producing the helical operative motion being realized by the tool itself Indeed, since the cutting edges of the tap are arranged on a helix with a pitch equal to the pitch of the thread being tapped, and the work envelopes the tap over its whole surface, the tap and work constitute a kinematic screw pair providing for helical motion F_r (R_1T_2) of the tap Consequently, the machine has a simple kinematic group instead

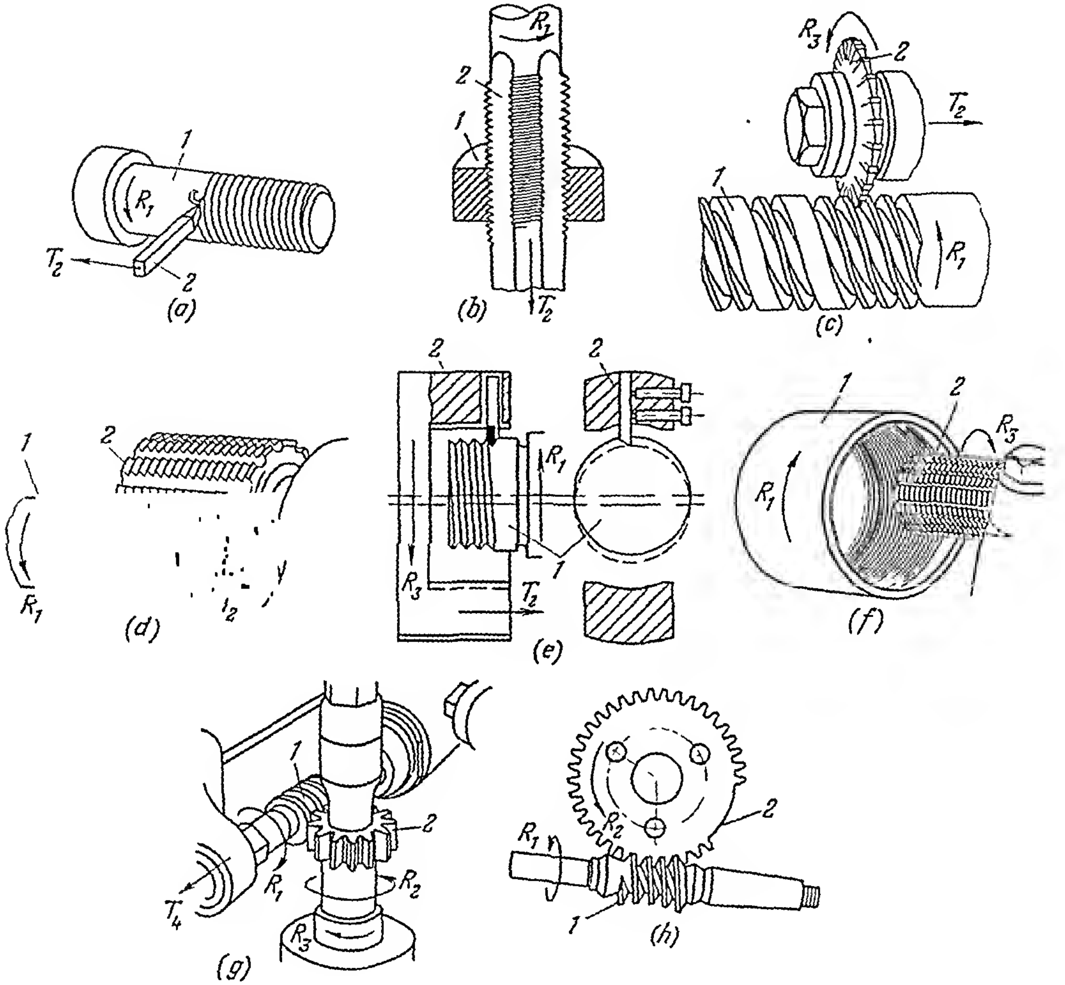

Fig. 30. Motions of the operative members in thread-making machines (*1*—work; *2*—cutting tool):

Cutting thread with: (*a*) single-point threading tool; (*b*) tap; (c) single-thread milling cutter; (*d*) multipl[illegible] ith annular threads; (*e*) rotary tool head with threading tools; (*f*) helical [illegible] cutter; (*g*) rotary cutter in generating a straight worm; (*h*) rotary cutter in generating a globoid (Hindley) worm

of a complex one. Thus, no setting-up is required to the pitch of thread to be cut; only setting-up device i_v remains to obtain the required cutting speed. No feed motion is available in such machine tools (having a single formative operative motion).

A helical surface can also be produced as one tangent to a series of auxiliary surfaces which are produced, in turn, by a separate rotary motion F_v (R_3) of the cutting tool (see Fig. 30*c*, *d* and *e*). The auxiliary surfaces are

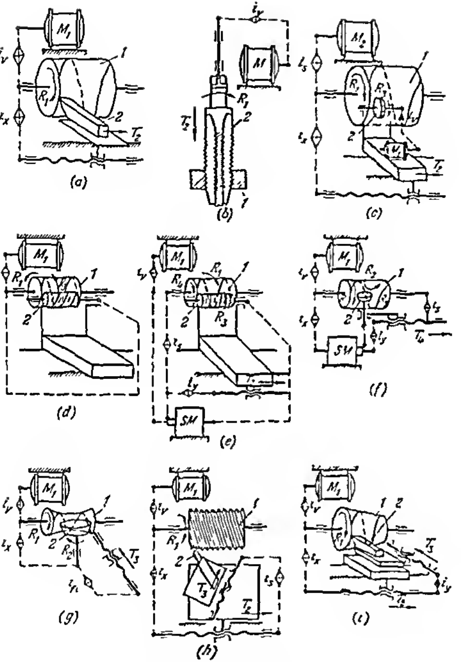

Fig. 31. Kinematic structures of thread cutting machines (1—work 2—cutting tool)

positioned in reference to the work by the helical motion F_s (R_1T_2) of the work and tool. The motion F_v (R_3) provides the required cutting speed, while motion F_s (R_1T_2) provides the feed. The same principle is employed in cutting threads with a single-thread milling cutter (Fig. 30*c*), multiple-thread milling cutter with annular threads (Fig. 30*d*), rotary tool head enveloping the work (Fig. 30*e*) and grinding wheels. In all of these cases, the kinematic structure of the machine consists of two kinematic groups: a complex group set up by the setting-up devices i_x and i_s, and a simple group with the setting-up device i_v (Fig. 31*c*).

If a helical (hob-type) thread-milling cutter is used in which the tooth profile, pitch and length of thread are the same as the profile, pitch and length of thread to be cut, the method of producing the helical surface remains, but the structure of the machine must be altered. The rotation of a helical thread-milling cutter (Fig. 30*f*) leads to movement of the cutting edge in reference to the work along the cutter axis (corresponding to the elementary motion T_2 in the preceding methods). The velocity of this motion must be co-ordinated with work rotation R_1 to obtain a thread of the specified pitch. To accomplish this, the machine must have an internal constraint between the work spindle and the cutter spindle (Fig. 31*d*) so as to produce the helical operative motion of the cutter edge. Here, two motions, one complex F_s (R_1T_2) and the other simple F_v (R_3), have been replaced by a single complex motion F_v (R_1R_3). Therefore, this machine possesses only a single complex kinematic group.

The internal constraint must provide for the following basic displacement:

1 revolution of the work $\longrightarrow P$ mm longitudinal travel of the cutting edge

Since the pitch of the cutter and that of the work are equal, the cutting edge must be displaced by one pitch per revolution of the cutter, i.e. the cutter must also make one revolution during one revolution of the work. No setting-up device for the thread pitch (for the path of the helical motion) exists in the internal constraints. The machine is set up to the required cutting speed by means of setting-up device i_v in the drive gear train. A machine tool based on this structure has no device for setting up the rate of feed. The number of auxiliary surfaces, to which the helical surface being produced is tangent in a length equal to one pitch, is equal to the number of teeth on the cutter. This is, of course, insufficient and therefore a slow helical motion F_s (T_2R_4) is added to the main motion F_v (R_1R_3). The number of auxiliary surfaces and, consequently, the finish of the surface being produced, are controlled by means of the added motion.

The structure of a machine tool of this type (Fig. 31*e*) consists of two complex kinematic groups, connected by a summation device and producing the motion F_v (R_1R_3) set up by device i_v and the motion F_s (T_2R_4) with

the setting up devices i_y and i_s. The basic displacement for setting up i_y is

$$1 \text{ revolution of the work} \rightarrow P \text{ mm longitudinal travel of the cutter}$$

where P is the pitch of the thread to be cut.

In addition to the methods previously mentioned, worms and lead screws are sometimes cut by the generating method.

To obtain the thread profile of a multiple-start worm by the generating method it is necessary to have a rotary cutter in the form of a toothed gear with a tooth profile which in generation is conjugate to the profile of the thread on the worm being cut (Fig. 20g). This gear-shaped cutter must roll without slipping along the worm being cut. If the number of teeth of the rotary cutter is denoted by z and the number of starts and module of the worm being cut by k and m respectively, then to obtain the rolling motion it is necessary for the cutter to make $\frac{L}{\pi m z}$ revolutions as it travels lengthwise over the distance L. To extend the profile of the worm over the whole helical surface a helical operative motion must be imparted to the cutting edge in addition to the previously mentioned motion. In respect to the second motion, the rotary cutter must travel longitudinally a distance equal to the thread pitch $P = k\pi m$ during the time the work makes one full revolution.

Hence, to cut the thread of a worm with a *rotary cutter* the machine must have two complex motions, each of which is produced by two elementary motions, one rotary and the other rectilinear. In the actual cutting process one of these motions provides the cutting speed and the other provides the feed. If two such complex motions are combined, the production capacity of this machining method will not be very high since a large number of passes will be required to cut the worm. This can be avoided by replacing either of these motions by another, also complex motion, but one composed of two elementary rotary motions. Then this motion will produce the cutting action, remaining continuously within the limits of the work length.

Let us replace the first motion—the rolling motion of the rotary cutter. The component motions are defined by the following basic displacements:

$$L \text{ mm longitudinal travel of the cutter} \rightarrow \frac{L}{\pi m z} \text{ revolutions of the cutter}$$

Then, since longitudinal travel of the cutter over the distance L mm corresponds to rotation of the worm blank through $\frac{L}{k\pi m}$ revolutions, we can write

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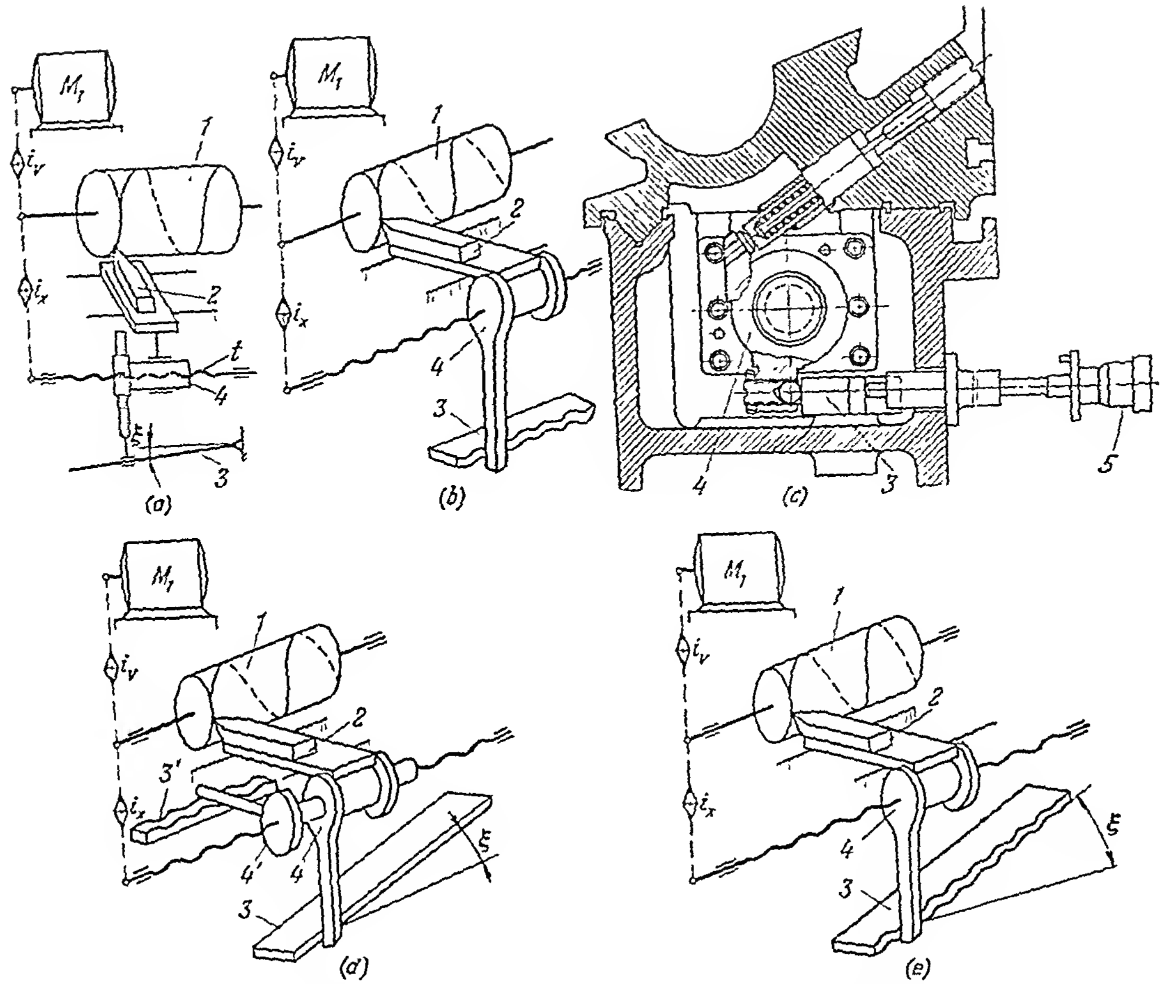


Fig. 32. Kinematic arrangements of thread-making machines having a correction bar: (a) for setting up to the thread pitch; (b) for correcting the thread pitch errors; (c) for setting up to the thread pitch (construction); (d) and (e) for setting up to the thread pitch and for correcting pitch errors

The value P' is selected near to the required pitch P and allowing change gears i_x to be easily calculated, so that the setting angle ξ of the correction bar will be small.

Illustrated in Fig. 32*b* is a machine having a correction bar which eliminates kinematic errors that occur in the thread-cutting gear train. The effective surface of the bar is of a shape (usually curvilinear) leading to nut rotation through small fractions of a revolution, if for some reason the nut is traversed during one revolution of the work over a distance

longer or shorter than the pitch of the thread to be cut. The bar is linked to the lead-screw nut through a rigid lever. This arrangement yields better results than the arrangement with a rack-and-pinion drive of Fig. 32a, since the latter is subject to backlash. The arrangement with a rigid lever is employed in practice. One construction of a nut, turned by a rigid lever which is actuated by a correction bar set at an angle, can be seen in Fig. 32c. Bar is set to the required angle by means of two micrometric screws with heads. A spring-loaded device holds the lever, rigidly secured to the nut, continuously against the correction bar. The lever contacts the bar through a hardened steel ball.

Attempts have been made to eliminate kinematic errors in the thread-cutting gear train and to obtain more accurate setting-up facilities by the provision of two correction bars 3 and 3 (Fig. 32d) or a single correction bar with a curvilinear contour (Fig. 32e). Neither arrangement has worked in practice. As a fact, additional errors are introduced by having two nuts turned by the bars in the first arrangement, and the accuracy of the whole system may even be lowered. When the bar in the second arrangement is set at an angle, the position of its contour is altered so that it can no longer effectively eliminate the kinematic errors of the gear train. Therefore systems with a single correction bar are most frequently employed. The purpose of this bar is either to eliminate kinematic errors or to set up the machine more accurately to the pitch of the thread being cut. In the latter case, the bar serves as a supplementary setting-up device to obtain the required path of the formative motion. Moreover, systems with a single correction bar are simpler than others from the manufacturing point of view.

A kinematic indexing group is introduced into the kinematic structure of machines designed to cut multiple-start threads. This does not apply to machines using a rotary cutter (thread generators) in which the indexing process is accomplished by the formative motion. The indexing motion may be either rotary or rectilinear depending upon the member involved—the work spindle or the carriage.

The corresponding basic displacements of the final members of the indexing gear train are

n_{ind} revolutions of the index plate $\to \frac{1}{k}$ revolution of the work

or

n_{ind} revolutions of the index plate $\to \frac{P}{k}$ mm longitudinal travel of the carriage

where k — number of starts of thread to be cut
P — lead of the thread to be cut

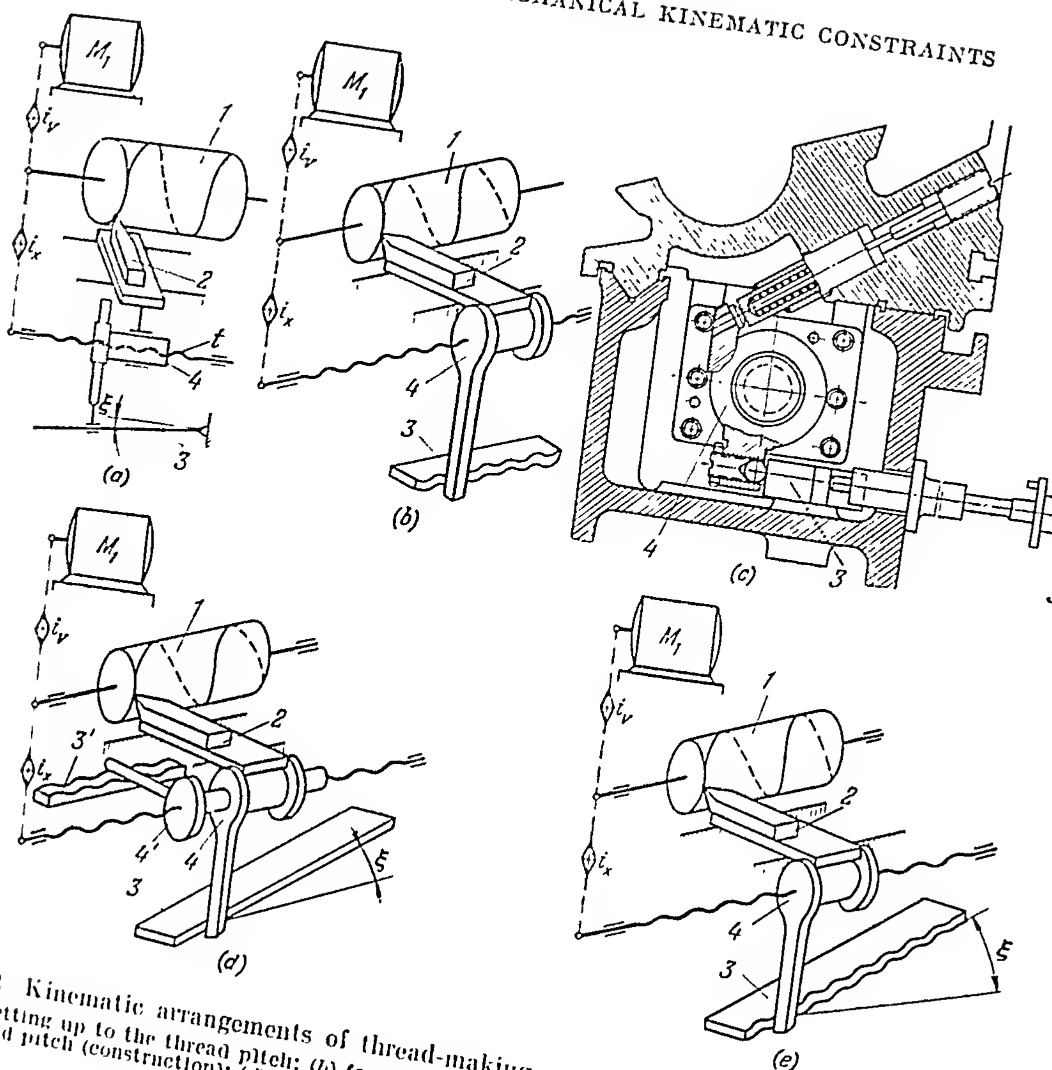


Fig. 32 Kinematic arrangements of thread-making machines having a correction bar: (a) for setting up to the thread pitch; (b) for correcting the thread pitch errors; (c) for setting up to the thread pitch (construction); (d) and (e) for setting up to the thread pitch and for correcting pitch errors

The value P' is selected near to the required pitch P and allowing change gears i_x to be easily calculated, so that the setting angle ξ of the correction bar will be small.

Illustrated in Fig. 32b is a machine having a correction bar which eliminates kinematic errors that occur in the thread-cutting gear train. The effective surface of the bar is of a shape (usually curvilinear) leading to nut rotation through small fractions of a revolution, if for some reason the nut is traversed during one revolution of the work over a distance

longer or shorter than the pitch of the thread to be cut The bar is linked to the lead screw nut through a rigid lever This arrangement yields better results than the arrangement with a rack and pinion drive of Fig 32*a* since the latter is subject to backlash The arrangement with a rigid lever is employed in practice One construction of a nut turned by a rigid lever which is actuated by a correction bar set at an angle, can be seen in Fig 32*c* Bar *3* is set to the required angle by means of two micrometric screws with heads *5* A spring loaded device holds the lever rigidly secured to the nut continuously against the correction bar The lever contacts the bar through a hardened steel ball

Attempts have been made to eliminate kinematic errors in the thread cutting gear train and to obtain more accurate setting up facilities by the provision of two correction bars *3* and *3* (Fig 32*d*) or a single correction bar with a curvilinear contour (Fig 32*e*) Neither arrangement has worked in practice As a fact additional errors are introduced by having two nuts turned by the bars in the first arrangement and the accuracy of the whole system may even be lowered When the bar in the second arrangement is set at an angle the position of its contour is altered so that it can no longer effectively eliminate the kinematic errors of the gear train Therefore systems with a single correction bar are most frequently employed The purpose of this bar is either to eliminate kinematic errors or to set up the machine more accurately to the pitch of the thread being cut In the latter case the bar serves as a supplementary setting up device to obtain the required path of the formative motion Moreover systems with a single correction bar are simpler than others from the manufacturing point of view

A kinematic indexing group is introduced into the kinematic structure of machines designed to cut multiple-start threads This does not apply to machines using a rotary cutter (thread generators) in which the indexing process is accomplished by the formative motion The indexing motion may be either rotary or rectilinear depending upon the member involved—the work spindle or the carriage

The corresponding basic displacements of the final members of the indexing gear train are

$$n_{nd} \text{ revolutions of the index plate} \rightarrow \frac{1}{k} \text{ revolution of the work}$$

or

$$n_{nd} \text{ revolutions of the index plate} \rightarrow \frac{P}{k} \text{ mm longitudinal travel of the carriage}$$

where k number of starts of thread to be cut
P lead of the thread to be cut

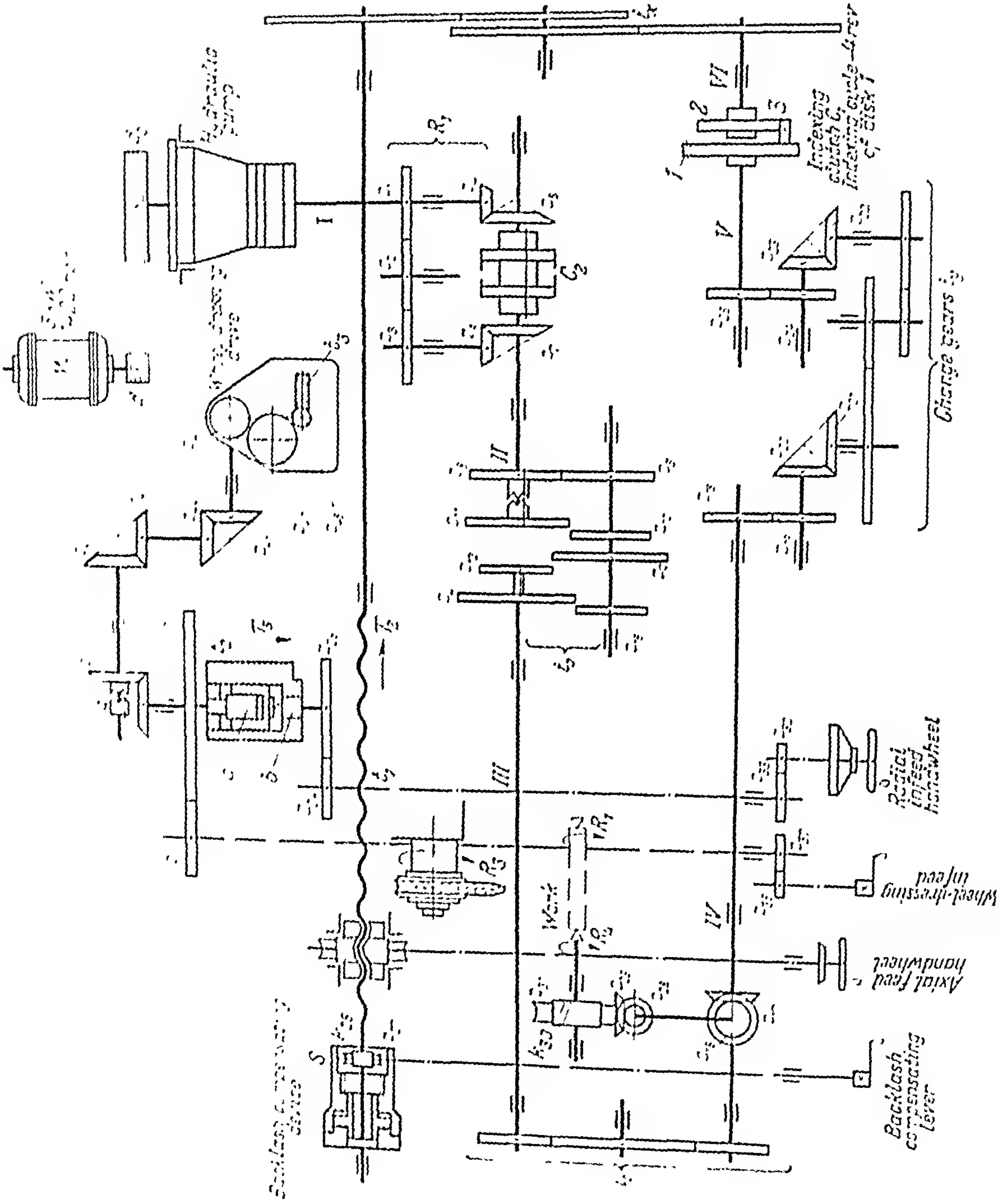


Fig. 34. Kinematic diagram of the worm grinding machine, model HSS33B, made by the Klingelnberg Co., FRG (structure class K23)

Further motion from this member will be through the internal constraint to the table and workpiece

The helical feed motion is a complex motion with an open path It must be set up to all five parameters to the path by means of the pitch change gears i_x to the velocity by the feed change gears i_s and feed gearbox i_0 to the direction by the reversing mechanism R and to the path length and initial point by dogs arranged on the table and actuating the hydraulic controls of the grinder The indexing motion group *Ind* (R_i) is simple in structure

The kinematic pair whose elements are the work spindle and the work head housing constitutes the internal constraint of this group Its external constraint passes from the same motor to the work spindle and then from member z_{17} to disk *1* in indexing clutch C_1 The indexing and feed groups are interconnected in series Clutch C_1 disengages the thread cutting gear train and then with the work table stationary, disk *1* of the indexing clutch makes four revolutions After this locking member *3* is extended to engage disk *2* The indexing change gears i_y arranged between disk *1* and the work are set up to the number of starts on the worm being ground The wheelhead is set in the radial direction by hand using stop *b* which is preset to the required value To dress the grinding wheel the wheelhead is fed in radially by an amount equal to the wheel wear by a screw with a pitch *t*.

Semiautomatic Thread Milling Machine, Model 5M5B62

This machine is intended for milling short threads of a diameter up to 100 mm with a multiple thread cutter The width of the cutter must be equal to or exceed the length of the thread to be milled The structure of this machine (Fig 3) consists of two formative kinematic groups I_v (R_1) and F_s (R_2T_3) The cutting motion group I_v (R_1) is of simple structure with speed change gears i_v and an electrical change over switch for revers ing cutter rotation

The *feed motion group* F_s (R_2T_3) is of complex structure it has one inter nal kinematic chain and a drive gear train The internal train is arranged between the work spindle and cam drum *1* A helical ring mounted on the drum sets up the miller to the pitch of the thread to be cut Motion in the internal train is powered from motor M_2 and is transmitted through feed change gears i_s to worm gearing $\frac{1}{4}$ and further through the housing of the differential and planet gears the lower central bevel gear and worm gearing $\frac{-}{1}$ to the work spindle Motor M_3 powers the auxiliary motion The radial feed in motion group *FI* (T_4) is simple powered by motor M_4 it

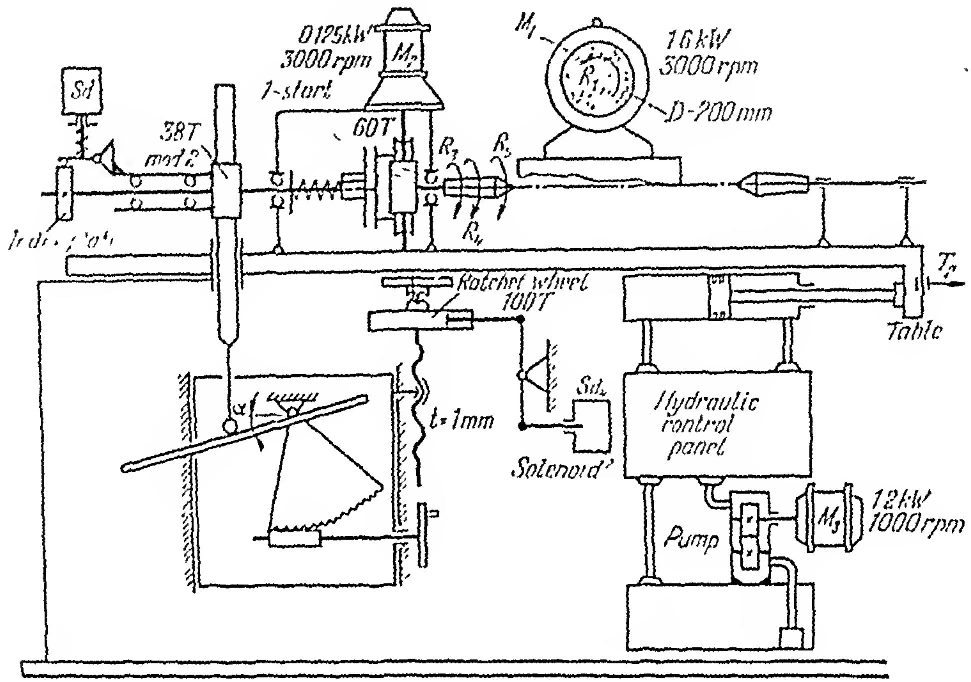


Fig. 36. Kinematic diagram of the semiautomatic hob sharpening machine, model 3A642 (structure class K23)

rack rotates a rack pinion. The latter, through locking member and index plate, rotates the work spindle. The motion $F_s\ (T_2R_3)$ is set up to the required path by setting the bar to the corresponding angle α; to the required path length by setting dogs on the table; and to the velocity and direction by making the necessary adjustments on the hydraulic control panel (not shown in the diagram).

The *indexing motion group Ind* (R_4) is of simple structure. It consists of a kinematic pair, whose elements are the work head and work spindle, and the drive train from the motor. It is evident from the diagram that the work spindle participates in both the formative and indexing motions. Therefore, the work spindle belongs to two kinematic groups. These groups are interconnected by the *consecutive indexing method*, in which the formative train is disengaged when the locking member, controlled by solenoid *Sd*, is retracted from the slot of the index plate. At this, the work spindle is rotated from motor M_2, this rotation being independent of table travel.

As to its path length parameter, the indexing motion is set up to the number of teeth on the hob to be ground by means of interchangeable index plates. The other parameters of this motion are constant.

The *kinematic group for the feed-in motion FI* (R_5) is also simple in structure. It consists of the kinematic pair—the work spindle and the work head—and a separate drive from solenoid Sd_4. Motion is transmitted from the solenoid through a rocker arm and a ratchet mechanism (100T) to a feed screw which moves the base of the bar. Motion is transmitted further through the slide block with the rack teeth to the work spindle. The feed-in group FI (R_5) is interconnected in parallel with the formative group F_2 (T_2R_3), the work spindle participating simultaneously in helical and rotary motions. As pointed out previously, this method of group interconnection requires a summation mechanism in the formative group. The base of the bar, in our case, acts as the summation device. In its motion, it imparts a supplementary displacement to the slide block and work spindle. The feed-in motion is set up to the path length parameter by adjusting the ratchet pawl to engage one, two or three teeth upon each stroke of solenoid Sd_4. The other parameters are constant.

The feed-in motion is periodic. It must set the work (hob) to a new depth of cut after z full strokes (back and forth) of the table.

The kinematic chains of the grinder are set up by setting the bar to the required angle α. The basic displacements and the kinematic balance equation for the hob lead train are

1 revolution of the hob $\rightarrow P$ mm of longitudinal travel of the hob

where P is the lead of the helical flutes on the hob being sharpened.

The kinematic balance equation is

$$1 \times \frac{2\pi \times 38}{\tan \alpha} = P$$

The setup formula is

$$\tan \alpha = \frac{76\pi}{P}$$

5-2. Kinematic Structure of Relieving Machines

Relieving Methods

The relieved surface behind the cutting edge of a hob tooth is a helical surface of a profile that coincides with the profile of the hob tooth extended in the direction of the length of the surface along a helix.

This relief surface is shaped by a form tool (Fig. 37a) by means of a single complex formative motion made up of three interrelated elementary motions R_1, T_2 and T_3, i.e. one rotary and two rectilinear motions. If a conical hob is to be relieved (Fig. 37b), the complex operative motion is made up of four elementary motions, the rectilinear motion T_5 being added. This motion is required to position the relieved surface along a helix on a conical surface.

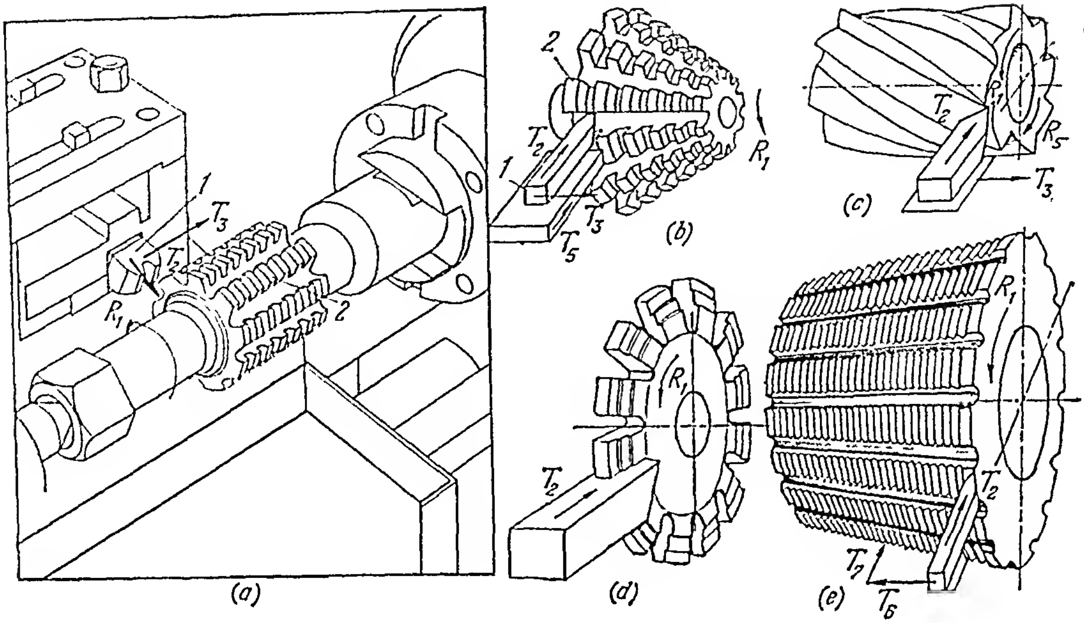


Fig. 37. Tool and work motions in relieving cutters:
(a) gear hob; (b) conical hob; (c) helical-flute plain milling cutter; (d) stocking gear milling cutter; (e) multiple-thread milling cutter

Evidently, the kinematic structure of lathes for relieving ordinary hobs constitutes a single kinematic group, accomplishing a single complex operative motion $F_v(R_1T_2T_3)$ for shaping the surface, but the internal kinematic constraint will consist, not of one, but of two or more internal kinematic chains.

If the internal constraint interconnects three elementary motions, it consists of two internal kinematic chains (Fig. 38*a*) with the setting-up devices i_x and i_y. The two chains have a common drive train with a setting-up device i_v. The internal chain with the setting-up device i_x provides for the operative motion of the cutting edge along an Archimedean spiral; chain i_y provides for motion along a helix.

If the group of the complex operative motion performs four elementary motions, as in relieving a conical hob (Fig. 38*b*), the internal constraint will consist of three internal kinematic chains.

As a rule, the reciprocating crosswise movement T_2 of the tool is effected by a cam (Fig. 39) since the path length of this motion is always small, rarely exceeding 10 mm.

In relieving a multiple-tooth cutter, it is necessary to accomplish the indexing process in addition to the formative motion. The teeth of a hob

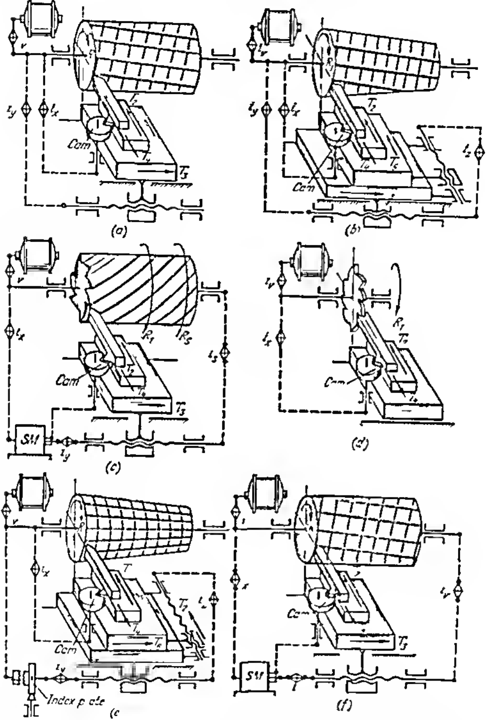


Fig 38 Structural diagrams of machines for relieving milling cutters and hobs
(a) gear hob (b) conical hob (c) helical flute plain milling cutter (d) disk type form milling cutter (e) tapered multiple-thread milling cutter (f) gear hob by a differential setup

Those of the pitch train (setting-up device i_y) are:

1 revolution of the work $\rightarrow \tau$ mm of longitudinal travel of the tool

The basic displacement of the cutting speed gear train (setting-up device i_v) are:

n_m rpm of the motor shaft $\rightarrow n_w$ rpm of the work

There is no feed motion, since the surface is shaped by a form tool whose cutting edge is of a shape coinciding with the profile of the tooth being relieved.

The kinematic structure of machines for relieving milling cutters of other types (not hobs) varies in accordance with the changes in the nature of the formative motions required.

Two formative motions, F_v (R_1T_2) and F_s (T_3R_5), are needed to relieve a helical-flute plain milling cutter (Fig. 37*c*), since the relieved surface is helical and is shaped by a single-point tool.

The motion F_v (R_1T_2) produces the curvilinear back of the tooth along an Archimedean spiral (Fig. 37*c*). The kinematic group accomplishing this motion consists of one internal kinematic chain with the setting-up device i_x (Fig. 38*c*) and one drive train—the cutting speed train i_v. The motion F_s (T_3R_5) produces the helix along the tooth length. This is a slow motion and constitutes the feed in the cutting process. It is accomplished by a kinematic group with an internal pitch gear train i_y and a drive train—the feed train i_s. Since the work spindle must execute two rotary motions simultaneously—rapid motion R_1 and slow motion R_5—the internal kinematic chains have a common branch and are interconnected by means of a summation mechanism (differential gearing).

In addition to the two formative motions, the machine also has an indexing motion. This is a simple rotary motion *Ind* (R_1) which is completed during the return crosswise motion T_4 of the tool. The kinematic group remains the same as before. During this time the relieving cam performs the supplementary function of an index plate. Consequently, the internal chain with the setting-up device i_x is called the indexing train.

The basic displacements for relieving a helical-flute plain milling cutter are:

for the indexing train (setting-up device i_x):

$$1 \text{ revolution of the cam} \rightarrow \frac{z}{k} \text{ revolutions of the work}$$

for the pitch, or lead, train (setting-up device i_y):

1 revolution of the work $\rightarrow P$ mm longitudinal travel of the tool

for the cutting speed train (setting-up device i_v):

n_m rpm of the motor shaft $\rightarrow n_w$ rpm of the work

for the feed train (setting-up device i_s):

1 revolution of the work $\rightarrow s$ mm of longitudinal travel of the tool

In relieving a stocking gear milling cutter, or any other type of form cutter with a form tool (see Fig 37*d*), no feed motion is required, and the kinematic structure of the machine consists of a single kinematic group accomplishing the formative operative motion F_v $(R_1 T_2)$ and the indexing motion *Ind* (R_1) as shown in Fig 38*d*

The structure of a machine for relieving a tapered multiple-thread milling cutter with a single-point threading tool (Fig 37*e*) consists of two groups (Fig 38*e*) (a) a kinematic forr ... in all the preceding machines, two operat ... to shape the Archimedean spiral and the ... (*b*) a kinematic indexing group, producing the second indexing (complex) motion Ind_2 $(T_6 T_7)$ which is required to dispose the threads along the width of the cutter on the tapered surface at distances from each other equal to the pitch τ. A separate indexing member, the index plate, is arranged in this group

The formative train with setting-up device i_x is set up in the same way as in the preceding case

The internal constraint of the indexing group accomplishing the complex motion Ind_2 $(T_6 T_7)$ consists of an internal kinematic chain with setting-up device i_z for traversing the tool ... ing train with setting up dev ... of the tool with rotation of the ... ments are to be obtained in setting up the change gears i_z

L mm of longitudinal travel of the tool $\rightarrow L \tan \alpha$ mm cross travel of the tool

where L = any arbitrary displacement of the tool

α = taper angle of the thread milling cutter to be relieved.

The basic displacements for the indexing train i_y are

n_{ind} revolutions of the index plate $\rightarrow \tau$ mm of longitudinal travel of the tool

A separate additional indexing group is required in a machine for relieving a hob, if the latter is of the multiple start type

Figure 38*f* shows the kinematic structure of a universal relieving machine for relieving a hob with a form tool by a differential setting-up method Here the constraint between the work spindle and the relieving cam has been accomplished through two internal kinematic chains by means of a summation mechanism (differential gearing)

The basic displacement 1 revolution of the work $\rightarrow \frac{z}{k}\left(1 + \frac{\tau}{P}\right)$ revolutions of the cam, can in this case be broken down into two

1 revolution of the work $\rightarrow \frac{z}{k}$ revolutions of the cam

and

$$1 \text{ revolution of the work} \longrightarrow \frac{z}{k} \times \frac{\tau}{P} \text{ revolutions of the cam}$$

Motion according to the first relationship is provided by a gear train with the setting-up device i_x; that of the second is provided by the gear train with the setting-up devices i_y and i_z. Now the internal constraint consists of three internal kinematic chains which are powered from a single drive train with the setting-up device i_τ. Since the shape of the surface produced by the generating cutting edge of the tool has not changed, the number of operative formative motions also remains constant, as does the number of formative kinematic motions. Only the structure of this group is changed. The increase in the number of setting-up devices simplifies the selection of change gears i_x. On the other hand, the summation mechanism reduces the kinematic accuracy of the machine.

As examples, we shall consider the setting-up calculations for two relieving machines.

Hob Relieving Machine, Model 1708 (Michigan Tool Co., USA)

This special nondifferential machine (Fig. 41) is intended mainly for relieving hobs in the mass or large-lot production of cutting tools. The machine is of rigid construction. This enables medium- and large-size hobs to be accurately relieved at a high rate of production.

The kinematic structure of the machine is made up of two kinematic groups: the cutting motion group F_τ $(R_1T_2T_3)$ and the indexing group Ind (R_1T_3).

The internal constraint of the cutting motion group consists of two internal kinematic chains, including the indexing train:

$$R_1 \longrightarrow \frac{120}{20} \longrightarrow i_x \longrightarrow \text{differential gearing} \longrightarrow \text{bevel gearing} \longrightarrow$$
$$\longrightarrow \text{interchangeable cam} \longrightarrow T_2$$

and the pitch train:

$$R_1 \longrightarrow \frac{84}{42} \longrightarrow i_y \longrightarrow \text{lead screw} \longrightarrow \text{carriage} \longrightarrow T_3$$

The external constraint consists of two motors and a differential with spur gears.

The indexing group consists of one internal kinematic chain—the lead train with the change gears i_y, providing for tool edge travel along the helical flutes on which the teeth of the hob are located.

The external constraint of this group is the same as that of the cutting motion group, except that a drive to the cam is added since the latter acts

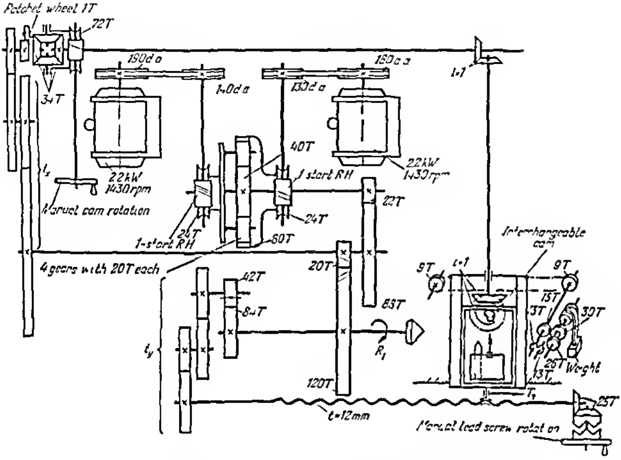


Fig. 41. Kinematic diagram of the hob relieving machine model 1708, made by the Michigan Tool Co., USA (structure class C13)

as a reversing device and an index plate. Thus, the indexing group is joined to the cutting motion group by the compound method of group interconnection. This principle is applied in all relieving machines.

Let us carry out the calculations for setting up this machine.

I. *Indexing train* (i_x)

1 revolution of the work $\rightarrow \frac{z}{k}\left(1 \pm \frac{\tau}{p}\right)$ revolutions of the cam

$$1 \times \frac{120}{20} \times i_x \times 1 \times 1 \times 1 = \frac{z}{k}\left(1 \pm \frac{\tau}{p}\right)$$

If $k=1$, then

$$i_x=\frac{z}{6}\left(1+\frac{\tau}{P}\right) \tag{12}$$

2. *Lead train* (i_y):

1 revolution of the work $\rightarrow \tau$ mm longitudinal travel of the tool

$$1\times\frac{84}{42}\times i_y\times 12=\tau$$

therefore

$$i_y=\frac{\tau}{24} \tag{13}$$

3. *Cutting speed train.* The work spindle drive consists of two electric motors, housed underneath in the base, and a differential with spur gears in the headstock. Four different spindle speeds are available: the motors operate separately, one at a time; they rotate together in the same direction; and they rotate together in different directions. The speed gearbox is controlled by electric change-over switches.

If only the right-hand electric motor is switched on, the spindle speed is

$$n_{rh}=1{,}430\times\frac{160}{130}\times\frac{1}{24}\times\frac{80}{40}\times\frac{22}{88}\times\frac{20}{120}\cong 6 \text{ rpm}$$

When only the left-hand motor is switched on, the spindle speed is

$$n_{lh}=1{,}430\times\frac{190}{140}\times\frac{1}{24}\times\frac{n_{40}}{n_{pc}}\times\frac{22}{88}\times\frac{20}{120}$$

where n_{40} and n_{pc} = speeds, rpm, of the sun gear ($40T$) and the planet carrier, respectively.

According to the Willis formula

$$\frac{n_{40}-n_{pc}}{n_{80}-n_{pc}}=-\frac{80}{40}=-2$$

If $n_{80}=0$, then $\frac{n_{40}}{n_{pc}}=3$.

Therefore

$$n_{lh}=1{,}430\times\frac{190}{140}\times\frac{1}{24}\times 3\times\frac{22}{88}\times\frac{20}{120}\cong 10 \text{ rpm}$$

If both motors operate simultaneously, then the spindle speeds are

$$n_{lh}+n_{rh}=16 \text{ rpm and } n_{lh}-n_{rh}=4 \text{ rpm}$$

Thus, the following spindle speeds are available: $n_1 = 4$ rpm, $n_2 = 6$ rpm, $n_3 = 10$ rpm and $n_4 = 16$ rpm. Reverse rotation is always at a speed of 16 rpm.

The indexing gear train includes differential gearing with a manual drive from a handwheel. This permits supplementary rotation to be transmitted to the cam without stopping the machine to change the initial point of the relieving motion.

A ratchet wheel on the left-hand central shaft of the differential gearing prevents rotation of the cam, and consequent crosswise motion of the relieving slide, upon the rapid return traverse of the carriage.

The carriage has a device that eliminates backlash in the thread of the lead screw and nut. This backlash develops during the relieving process. The threads of the nut and lead screw are held in contact by a weight acting through a gear, sprocket $9T$ and a chain linked to the carriage. The sliding gears $26T$ and $30T$ keep the nut in contact with the screw during carriage traverse in either direction.

The setting up procedure is complicated in this machine since the values of i_x and i_y in change gear calculations are expressed in decimals. Moreover, the setup formula for the indexing change gears depends upon three parameters z, τ and P. Since a large number of combinations of these three parameters is possible, this feature does not facilitate setting up procedure. General-purpose machine tools must have a kinematic structure that can be conveniently set up. Hence the change gears of these machines must have setup formulas based upon a single parameter. This can be done if the machine has a large number of correspondingly arranged setting up devices, and if differential gearing is introduced into the kinematic structure. In this connection we shall now consider the possible structural arrangements of relieving machines.

Figure 42a illustrates the kinematic structure of a nondifferential relieving machine consisting of a single formative group. If a hob is relieved with a form tool, the setup of the indexing change gears i_x, as mentioned above, is a function of three parameters $i_x = f_1(z, \tau, P)$, where z = number of teeth being relieved, τ = axial pitch of the hob thread and P = lead of the helical flutes.

The basic displacements of the indexing gear train are

$$1 \text{ revolution of the work} \rightarrow z\left(1 + \frac{\tau}{P}\right) \text{ revolutions of the cam}$$

If this sum of revolutions of the cam is linked with the revolutions of the work, not through one gear train but through two trains and a summation mechanism, we obtain a new kinematic structure (Fig. 42b). The relationship indicated above is provided for by two change gear units i_x and i_z. However, in this arrangement of change gears i_z, when this unit is driven from the spindle, its setup formula is again a function of the same three parameters. If change gear unit i_z is to be driven from the lead screw (Fig. 42c), then the setup formula of change gears i_z will depend upon only two param-

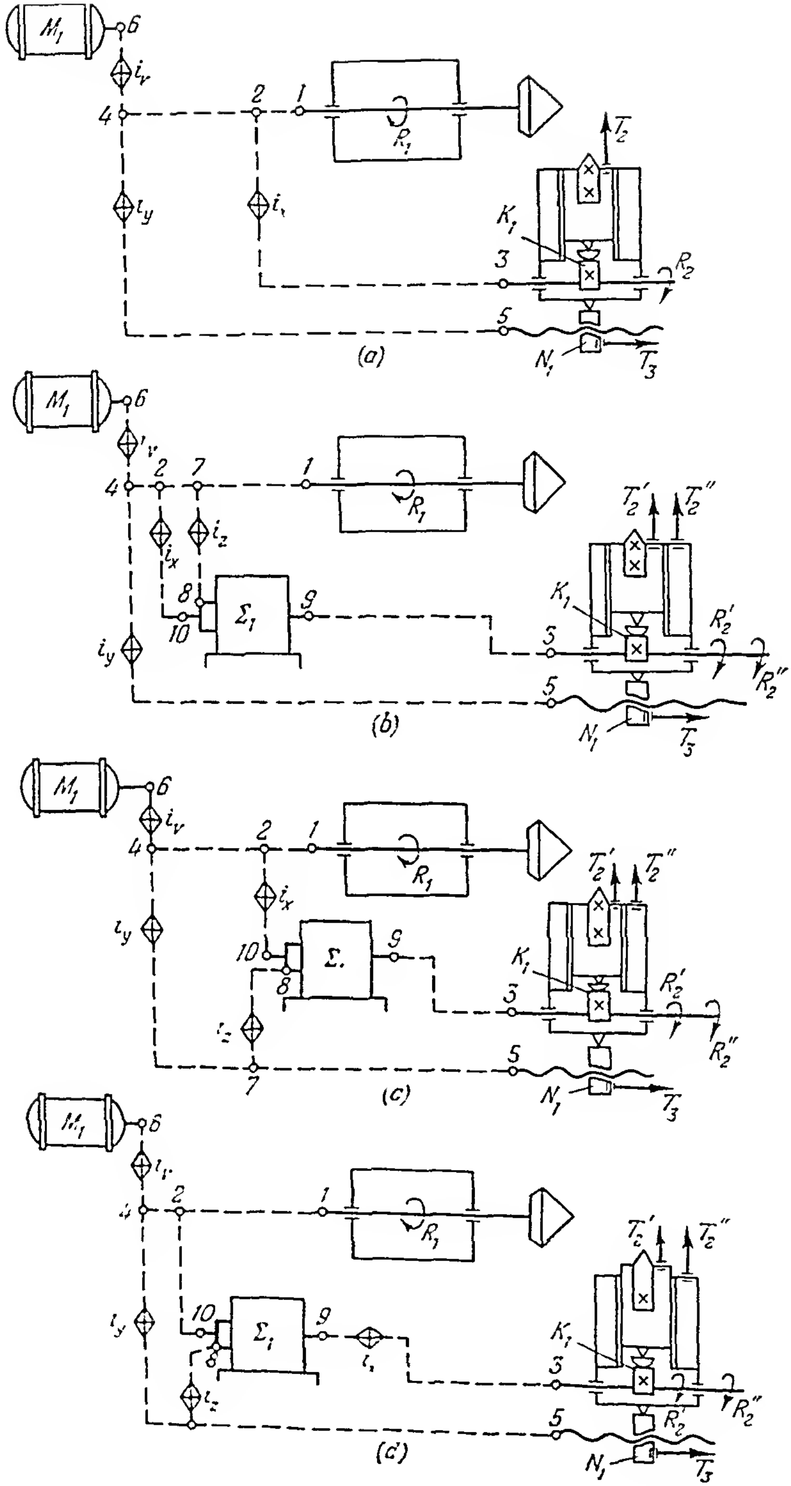


Fig. 42. Structural arrangements of relieving machines

eters (parameter τ is cancelled). Finally, if the indexing change gears i_x are transferred from section *2-10* to section *9-3*, following the differential gearing (Fig. *42d*), then each change gear unit will depend upon a single parameter. Thus

$$i_x = f_1(z) \qquad i_z = f_2(P) \quad \text{and} \quad i_y = f_3(\tau)$$

This last structure (Fig. *42d*) conforms to the structure of the majority of thread grinders having a relieving mechanism.

Owing to the substantial variable loads acting on the cam, the structures of general-purpose relieving machines are based on the arrangement shown in Fig. *42c*.

Next, we shall consider the kinematic structure and setting up procedure of one of the general-purpose relieving machines.

Semiautomatic Relieving Lathe, Model 1811

The model 1811 general-purpose lathe can be employed to relieve cutting tools of all types and to perform regular lathe operations. Hence, the kinematic structure of the lathe (Fig. 43) consists of a series of separate structures employed in relieving teeth of various shapes. The carriage has a second drive used for lathe operations and rough relieving. Here, motion is transmitted through the feed gearbox, feed rod, apron and a pinion-and rack drive. The maximum diameter of work that can be relieved is 240 mm, the maximum diameter of work turned in a chuck is 520 mm and the maximum distance between centres is 710 mm.

We shall consider the part of the structure (Fig. *43a*) used for relieving tools of the most complex shape, i.e. gear hobs.

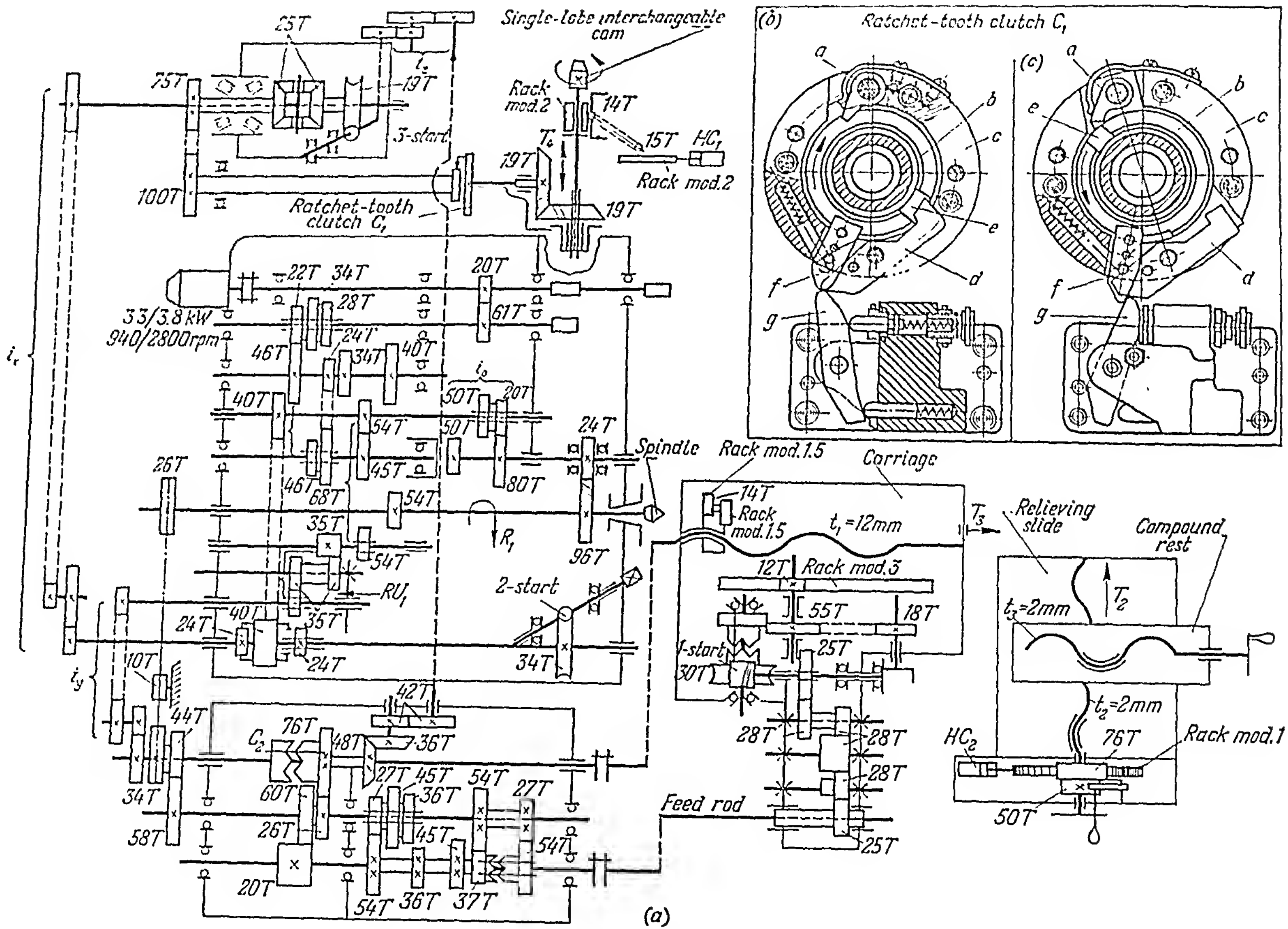
As mentioned above, the lathe is of the general-purpose type and differential gearing has been provided in the structure of its formative part to facilitate setting up. Therefore, the internal constraint of the motion group $\Gamma_\tau (B_1T_2T_3)$ consists not of two but of three internal kinematic chains. The first pitch gear train links the spindle and lead screw through change gears i_y. The second train—the indexing gear train—is arranged between the spindle and the cam, and passes from

i_0, gear *40T*, indexing change gears i_x

third gear train is also arranged betwee

through the pitch change gears i_y, clutch C_2, gearing $\frac{48}{36} > \frac{42}{42}$, differential change gears i_z and the differential gearing.

The external constraint of the cutting speed motion passes from the first speed step of the electric motor through the feed gearbox which provides 12 spindle speeds in a range from 2.8 to 63 rpm. The second speed step of the motor is switched on for return traverse of the carriage and for regular lathe


Single-lobe interchangeable cam
Rack mod.2
14T
15T
HC_1
Rack mod.2
19T
T_4
19T
25T
75T
i_x
19T
3-start
100T
Ratchet-tooth clutch C_1
22T
34T
28T
20T
61T
3.3/3.8 kW
940/2800 rpm
46T
24T
34T
40T
i_0
50T
20T
40T
54T
50T
24T
Spindle
46T
68T
45T
80T
26T
54T
R_1
96T
35T
54T
RU_1
2-start
40T
35T
24T
24T
34T
i_y
10T
44T
42T
76T
C_2
48T
36T
34T
60T
27T
45T
36T
54T
27T
58T
26T
45T
54T
20T
54T
36T
37T
(a)
(b)
Ratchet-tooth clutch C_1
a
b
c
d
e
f
g
(c)
a
b
c
d
e
f
g
Rack mod.1.5
Carriage
14T
Rack mod.1.5
t_1=12mm
T_3
12T
Rack mod.3
55T
18T
1-start
30T
25T
28T
28T
28T
25T
Feed rod
Relieving slide
Compound rest
T_2
t_3=2mm
t_2=2mm
HC_2
76T
Rack mod.1
50T

operations This provides for three more spindle speeds, 95, 135 and 189 rpm, obtained through the feed gearbox

This kinematic group has a number of devices that extend the processing capacity of the lathe, for instance, a ratchet tooth clutch C_1 (Fig 43a) This clutch has a driving pawl *a* (Fig 43*b* and *c*) by means of which drive shaft *b* rotates driven shaft *c* during working travel of the carriage When the direction of rotation of drive shaft *b* is reversed, the pawl *a* is disengaged and the shaft *c* stops This is necessary so that the cam does not rotate during the rapid return travel of the carriage Such cam rotation would lead to extremely rapid crosswise movements of the relieving slide that could result in a breakdown

In the given lathe, the construction of this clutch has been further developed so that the relieving slide always stops in its initial retracted position To this end, the driven shaft *c* carries the swinging stop *d* Upon the reversal of drive shaft *b*, its slot *e* engages stop *d* (Fig 43c), and the driven shaft is turned to the position where positive stop *f* runs against stop *g* At this, the relieving slide is in its extreme retracted position To withdraw the tool completely from the [illegible] has a tapered surface, in addition [illegible] This tapered surface is in contact [illegible] when the cam is lowered by a hydraulic cylinder during the return travel of the carriage

A coarse pitch unit is incorporated in the design to enable multiple start hobs to be relieved in cases when the hob thread lead is several times larger than the lead screw pitch Under ordinary conditions such leads would require gearing up in change gears i_y, which is undesirable The coarse-pitch unit includes sliding gear 54*T*, mounted on a shaft shown in the kinematic diagram (Fig 43a) adjacent to the spindle and slightly below it The diagram shows the left position of this gear (when the coarse pitch unit is engaged) Here the lead screw is driven, not directly from the spindle, but through countergearing $i_0 = \frac{96}{24} \times \frac{50}{50} = 4$ or $i_0 = \frac{96}{24} \times \frac{80}{20} = 16$

In this case, change gears i_y will be set up with reduction gearing again

The pitch gear train includes reversing unit RU_1 and sliding gear 35*T* which are used in relieving right and left hand hobs

A wide face gear 40*T* is provided in the indexing gear train after the reversing unit RU_1 This gear has two additional internal gear rings with 24 teeth each If gear 40*T* is shifted to its right hand position it is possible to turn the spindle by hand without rotation of the cam This is needed in setting up the beginning of the relieving motion and in indexing from start to start

The semiautomatic operating cycle is accomplished with the aid of various electrical and hydraulic devices the tool being set automatically to the depth of cut for relieving by means of hydraulic cylinder HC_2

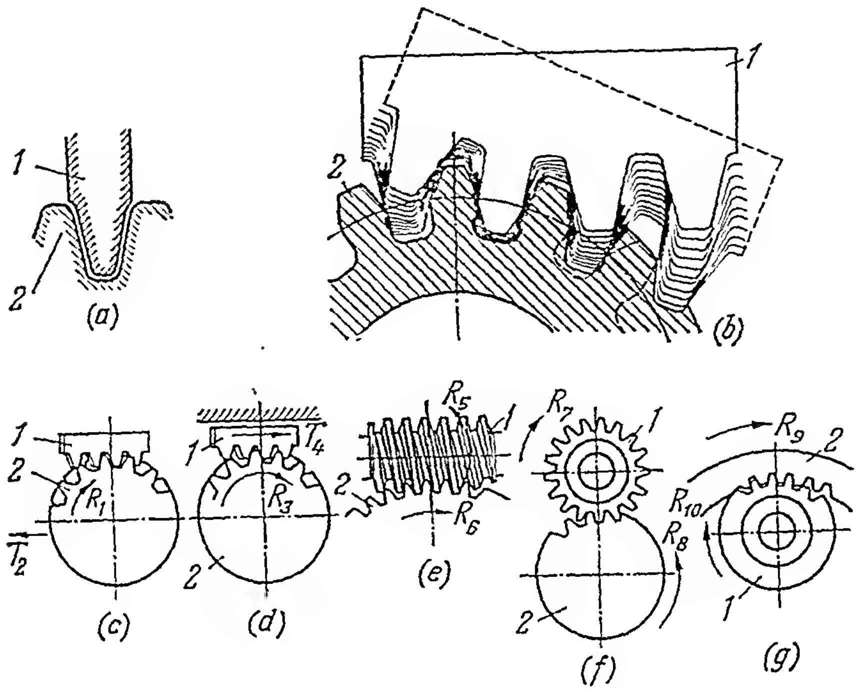


Fig. 44. Methods of producing the involute profile of spur, helical and herringbone gears: 1—cutting tool; 2—gear blank

tooth profile is a complex rolling motion, accomplished in machine tools by two elementary interrelated motions. If the basic rack becomes a cutting rack (rack-type cutter), the rolling motion is composed of motions R_1 and T_2 of the gear blank (Fig. 44*c*), or motion R_3 of the blank and motion T_4 of the rack-type cutter (Fig. 44*d*).

The version of relative motion of the rack-type cutter rolling about a stationary blank (Fig. 44*b*) is not applied in practice since it would unnecessarily complicate the construction of the gear-cutting machine.

If basic racks are arranged on a cylinder in such a manner that the generating contours are located along a helix, the rack-type cutter becomes a gear hob, and the operative motion for shaping the tooth is made up of two rotary motions R_5 and R_6 (Fig. 44*e*).

In addition to the tooth of a rack, the tooth of a spur or helical gear can also be employed as the generating contour. Then the operative motion used to produce the profile is made up of two interrelated rotary motions: R_7 and R_8 (Fig. 44*f*) for cutting external gears, and R_9 and R_{10} (Fig. 44*g*) for cutting internal gears.

Besides the operative motion required to shape the tooth profile, another operative motion of the generating contour is needed to extend the shape of the tooth along its full length. This motion may be simple rectilinear (for a spur tooth), or complex helical (for cutting a helical gear). In some cases, two formative motions are used for this purpose (in milling and grinding the tooth).

Based upon an analysis of the methods used to shape the teeth of spur, helical and herringbone gears, the following can be presumed.

1. *Machine tools cutting spur gears by the forming method must have* a structure with simple formative kinematic groups that produce formative motions only with the purpose of extending the shape of the tooth along its length. The tooth profile is produced by the cutting tool itself, a separate indexing motion being required in almost all cases. The main drive motion may be either rotary or rectilinear. The maximum number of formative kinematic groups is two.

2. Machine tools cutting spur gears by the generating method must have a structure consisting of complex and simple kinematic groups. A complex motion is required to shape the tooth profile. In most cases, these machines have no separate indexing motion. The main drive motion may be either simple or complex. The maximum number of formative kinematic groups is three.

The kinematic structures and setting-up procedures for the principal models of Soviet machine tools designed for cutting spur, helical and herringbone gears are considered in the following.

Machines which cut spur gears with disk type or end mill type gear milling cutters by the forming and tangent methods, intended for roughing purposes, are not to be considered since a gear grinder which grinds gears with a disk-type formed wheel has much the same structure. The kinematic diagram of such a grinder is illustrated in Fig. 57.

Gear-Hobbing Machine, Model 5E32

The universal gear hobbing machine, model 5E32 (Fig. 45), can cut various types of gears and, in particular, spur and helical gears up to 800 mm in diameter with a module up to 8 mm (of cast iron). The kinematic structure of this hobber is designed on the basis of the model 5Д32 gear hobber, previously manufactured by the Komsomolets Plant of Egoryevsk, but certain slight changes have been made in the new model. The whole kinematic structure is never employed simultaneously, it operates in parts in accordance with the shape of gear to be cut.

Let us examine these structures.

In cutting helical gears two formative kinematic groups of the hobber are employed. The cutting motion group F_v (R_1R_2) consists of a single

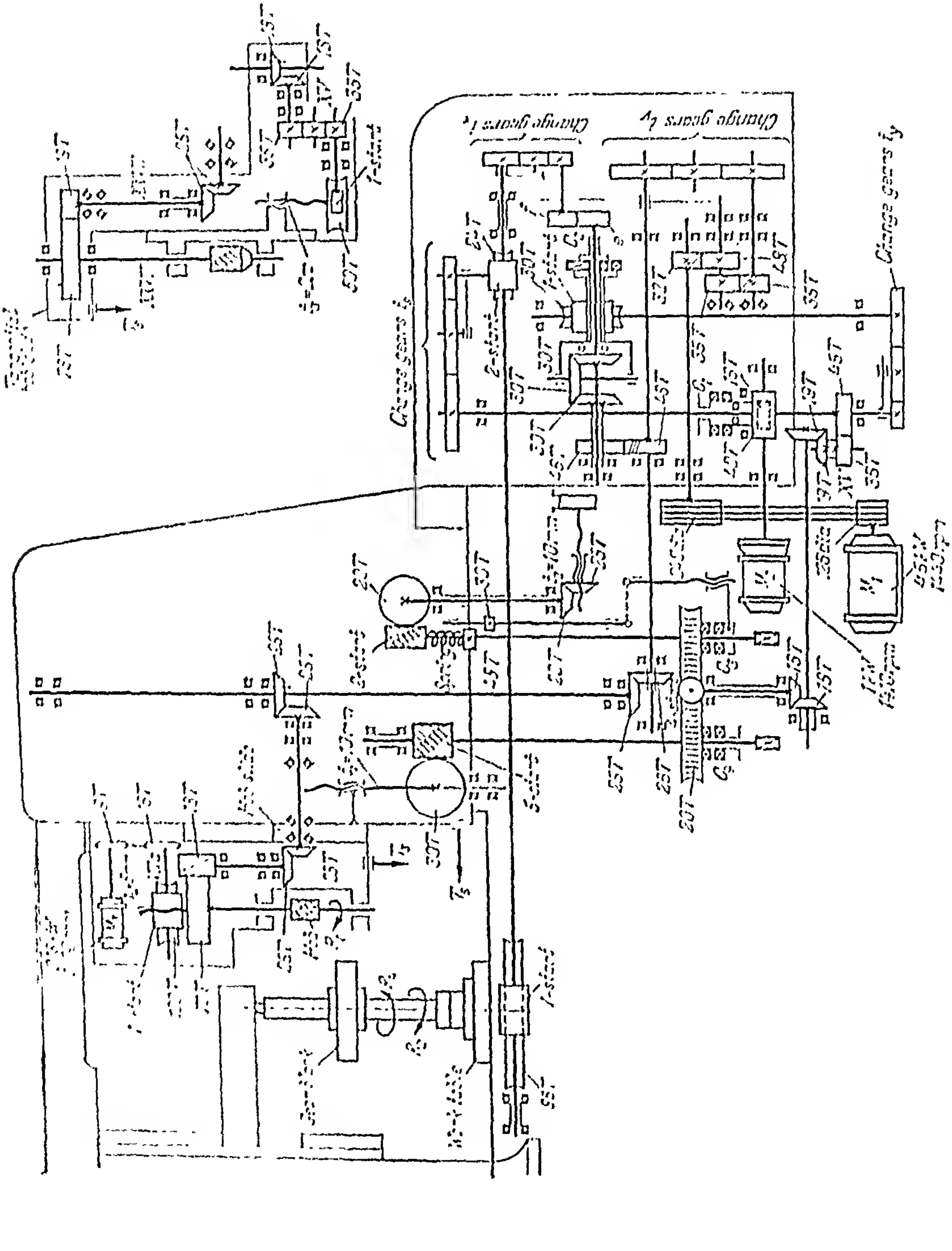


Fig. 45. Kinematic diagram of the gear-hobbing machine, model 5E32, made by the Komsomolets Plant of Egoryevsk (structure class C24)

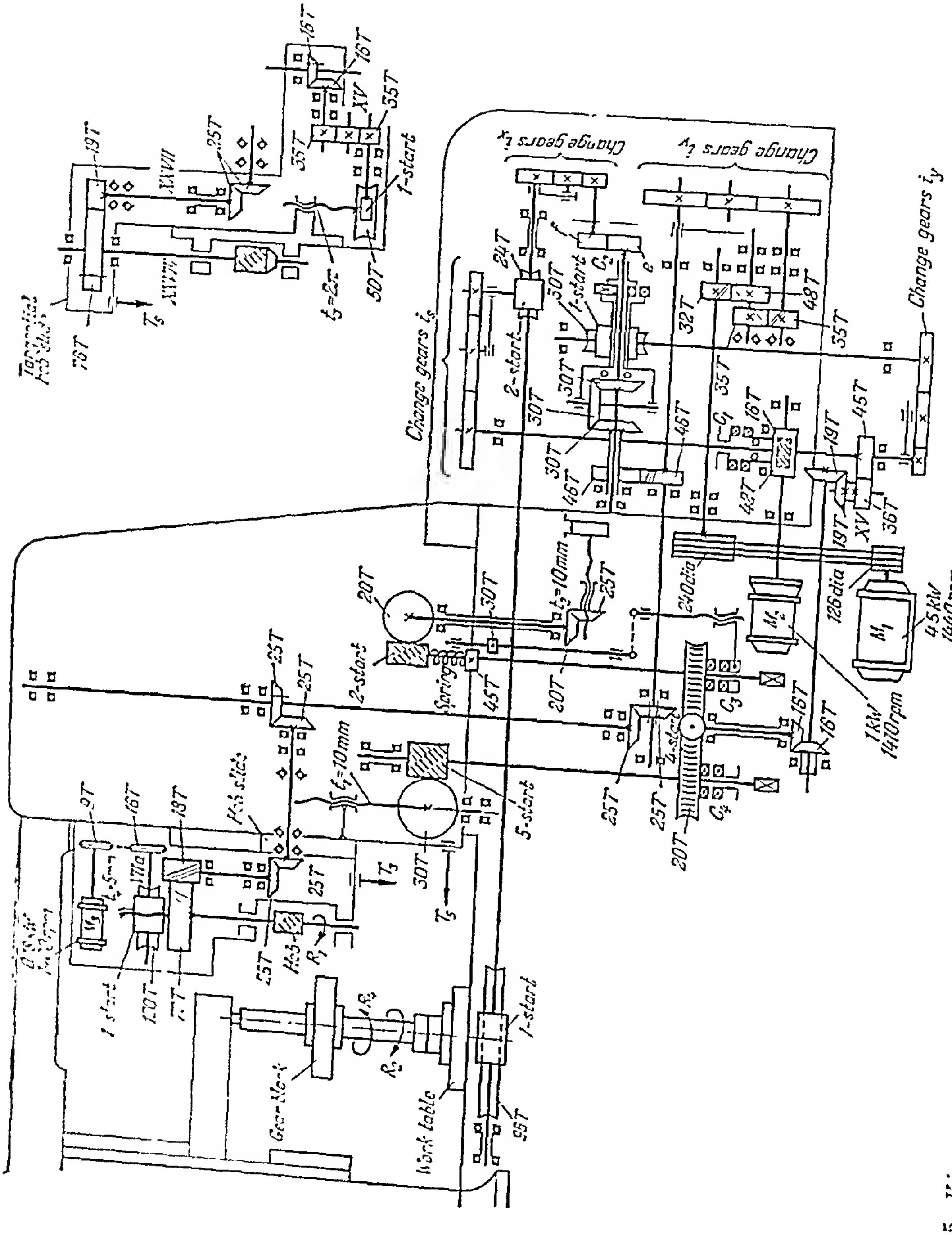
kinematic chain between the hob spindle and the table, passing through the differential and change gear i_x, and the drive train from motor M_1 transmitting motion through V-belts, helical gearing and change gears i_v (motion is transmitted further through the internal train to the hob spindle and the table with the gear blank). Motion F_v is set up as to its path by change gears i_x, and as to velocity and direction, by change gears i_v.

The second formative kinematic group F_s $(T_3 R_4)$ has an internal train between the hob slide and the work table. The kinematic constraint between these units passes from the nut and vertical feed screw through two worm and two bevel gearing units, spur gears $\frac{36}{45}$, differential change gears i_y, worm gearing, differential planet carrier, spur gears $\frac{e}{f}$, change gears i_x and the worm gearing $\frac{1}{96}$ of the work table. The drive train transmits motion to this internal train from motor M_1 through change gears i_v, sun bevel gears of the differential, change gears i_x, worm gearing $\frac{2}{24}$, feed change gears i_s, engaged clutch C_1, and further through the internal train to the feed screw and work table. To provide rapid approach of the hob and its rapid withdrawal to the initial position, the internal train of motion F_s is powered by motor M_2 with clutch C_1 disengaged.

The feed motion F_s is set up in respect to all five parameters: differential change gears i_y set up the path of the motion; feed change gears i_s set up the velocity and direction; and adjustable dogs set up the path length and initial point. These adjustable dogs are located on the hob slide and they operate limit switches which switch off the main drive motor. The hob is set to its vertical initial position by hand through a shaft carrying the worm of the worm gearing $\frac{2}{20}$.

In cutting a spur gear, the kinematic group of the cutting motion F_v remains the same, except that the differential is excluded from its internal train (indexing gear train) to increase the operating accuracy of this train. No differential is required in cutting a spur gear

The differential is excluded from the kinematic arrangement by means of clutch C_2 which disengages the worm wheel from the planet carrier, engaging the latter to the shaft of the right-hand bevel sun gear of the differential. In this position, the bevel gears of the differential are locked, and the left input and right output shafts of the differential rotate together with the carrier (housing) as an integral shaft For this purpose, the hobber is furnished with two interchangeable jaw clutches, one with narrow and the other with wide jaws The wide-jaw clutch is installed for cutting spur gears. In this case, the feed motion F_s (T_3) becomes a simple motion



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The standard hob slide is removed and a tangential, or traversing, hob slide is installed (its diagram is shown separately in Fig. 45) if worm wheels are to be cut by the tangential-feed method or with a fly cutter. In all other features, the structure remains the same as in cutting helical gears.

In cutting a worm wheel by the radial infeed method, two kinematic groups are employed: the previously considered cutting motion group F_v and the radial infeed group FI (T_5) for traversing the hob slide stanchion. The internal constraint of this group is made up of the translatory kinematic pair formed by the stanchion and base.

The external constraint goes from motor M_1, through the speed gear train, change gears i_v, sun gear of the differential, feed change gears i_s, gearing, worm gearing $\frac{4}{20}$, worm gearing $\frac{2}{20}$, and bevel gearing $\frac{20}{25}$ to the horizontal feed screw with a pitch of $t = 10$ mm. This gear train incorporates clutch C_3 which automatically disengages stanchion infeed when the hob reaches the preset centre-to-centre distance between the hob and the gear blank.

After a certain length of travel, the stanchion runs up against a positive stop mounted on the base. When the stanchion stops, the worm wheel in the worm gearing $\frac{2}{20}$ also stops rotating. Then its mating worm, which continues to rotate, will travel axially as a rotating screw mating with a stationary nut, until gear $45T$ slides into mesh with gear $30T$. The latter begins to rotate and, with it, a screw whose nut shifts a fork that disengages clutch C_3. Thus the infeed motion ceases at a given amount of radial infeed. Clutch C_4 engages and disengages the vertical feed.

The hobber is set up for performing its principal operations in accordance with the setup formulas derived below.

1. Setting up the hobber for cutting helical gears with a hob.

Cutting motion $F_v(R_1R_2)$.

(a) *Indexing gear train* (*change gears* i_x)

The basic displacements are

$$1 \text{ revolution of the hob} \rightarrow 1 \times \frac{k}{z} \text{ revolutions of the gear blank}$$

where k = number of starts on the hob.

Multiple-start hobs are used, as a rule, only for roughing gears.

The kinematic balance equation is

$$1 \times \frac{72}{18} \times \frac{25}{25} \times \frac{25}{25} \times \frac{25}{25} \times \frac{46}{46} \times 1 \times \frac{e}{f} \times i_x \times \frac{1}{96} = \frac{k}{z}$$

and the setup formula is

$$i_x = 24 \times \frac{f}{e} \times \frac{k}{z} \qquad (14)$$

The single-pair change gears $\frac{e}{f}$ with fixed axes can have two ratios in the given hobber $\frac{e}{f} = 1$ and $\frac{e}{f} = \frac{1}{2}$. Its purpose is to extend the range of the number of teeth cut on gears. The ratio $\frac{e}{f} = 1$ is used for cutting gears having less than 161 teeth.

(b) *Cutting speed gear train (change gears i_v)*

1,440 rpm of electric motor $M_1 \longrightarrow n_h$ rpm of the hob

$$1\,440 \times \frac{126}{\ldots 0} \times \frac{32}{48} \times \frac{35}{35} \times i_v \times \frac{25}{25} \times \frac{25}{25} \times \frac{25}{25} \times \frac{18}{72} = n_h$$

and

$$i_v = \frac{n_h}{1\ldots 6} \tag{15}$$

Feed motion $F_v(T_3R_4)$

(c) *Differential gear train (change gears i_y)*

1 revolution of the gear blank $\longrightarrow P$ mm vertical travel of the hob

where $P = \frac{\pi m_n}{\sin \beta}$ mm = lead of the tooth helix

β = helix angle of the gear teeth

m_n — normal module

The kinematic balance equation is

$$1 \times \frac{96}{1} \times \frac{1}{i_x} \times \frac{f}{e} \times \frac{1}{\ldots} \times \frac{30}{1} \times \frac{1}{i_y} \times \frac{25}{36} \times \frac{19}{19} \times \frac{16}{16} \times \frac{4}{20} \times \frac{5}{30} \times 10 = P$$

and the setup formula is

$$i_y = \frac{25}{\pi} \times \frac{\sin \beta}{m_n k} = \frac{7.9577 \sin \beta}{m_n k} \tag{16}$$

(d) *Feed gear train (change gears i_s)*

1 revolution of the gear blank $\longrightarrow s_v$ mm vertical travel of the hob

$$1 \times \frac{96}{1} \times \frac{2}{24} \times i_s \times \frac{4}{36} \times \frac{19}{19} \times \frac{16}{16} \times \frac{4}{20} \times \frac{5}{30} \times 10 = s_v$$

and

$$i_s = \frac{3}{10} s_v \tag{17}$$

2. Setting up the hobber for cutting worm wheels with a hob by the radial infeed method

Cutting motion $F_v(R_1R_2)$

Set up in the same way as above

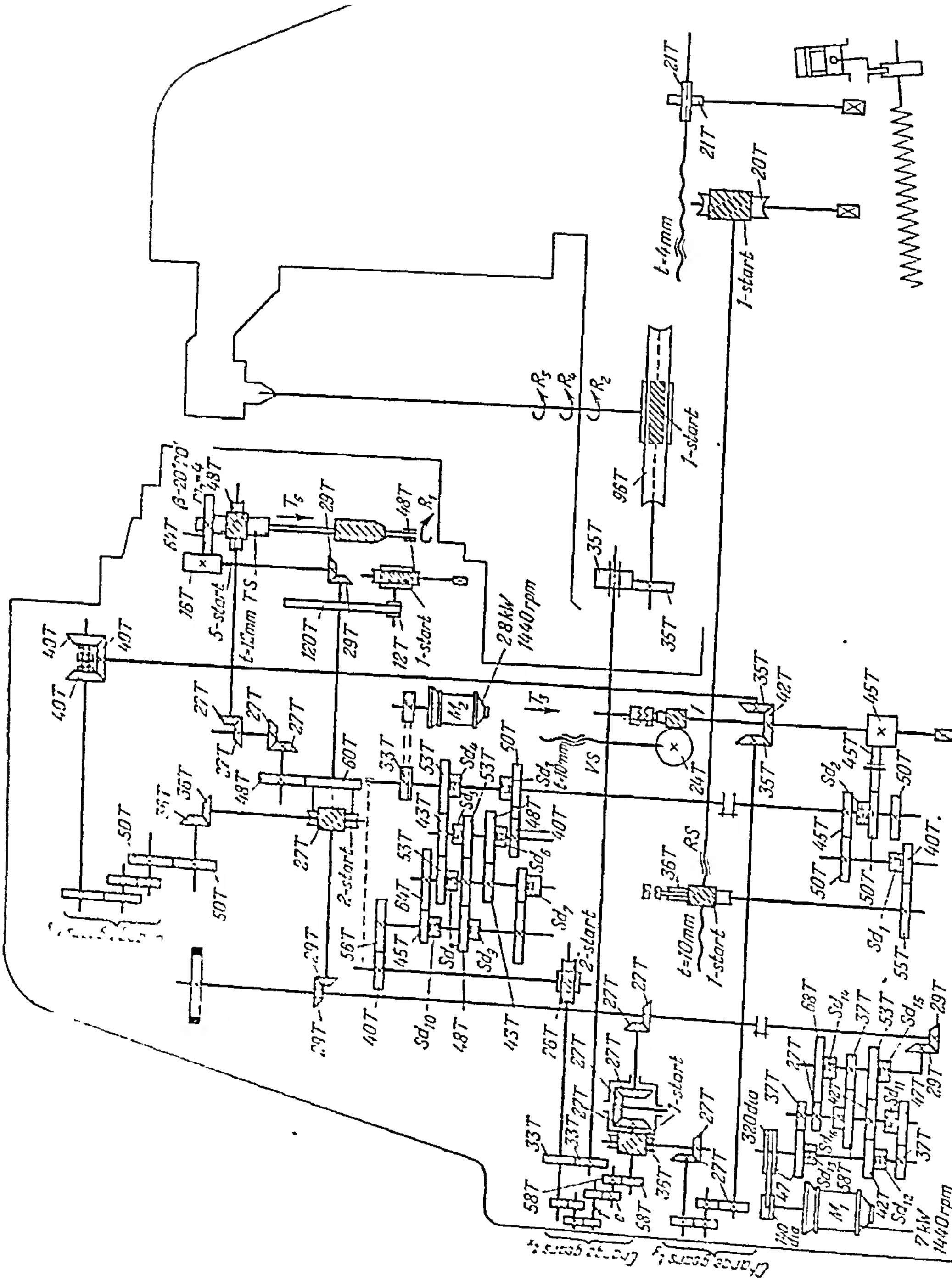

Fig. 46. Kinematic diagram of the semiautomatic hobbing machine, model 5K324, made by the Komsomolets Plant of Egoryevsk (structure class C24)

the hob spindle and work table through the sun bevel gear of the differential and change gears i_x, and an external constraint through which motion is transmitted from motor M_1 to the internal kinematic chain The cutting motion F_v (R_1R_2) is set up in respect to two parameters change gears i_x set up the path of the motion, and the speed gearbox, providing nine hob spindle speeds in a range from 5 to 310 rpm, sets up the velocity of the motion

The second kinematic group—the first feed group that produces the helical motion F_{s1} (T_3R_4)—consists of an internal kinematic constraint, also in the form of a single kinematic chain, interconnecting the vertical feed screw *VS* with the work table through the differential and change gears i_y The external constraint of this group, in the form of a drive gear train, transmits motion from motor M_1 into the internal kinematic chain, being linked to the latter through bevel gear $42T$ meshing with two bevel gears $35T$ Motion F_{s1} is to be set up in respect to all five parameters change gears i_y set up the path (to the lead of the tooth helix), adjustable dogs on the hob slide, disengaging this motion, set up the path length and initial point, a nine step feed gearbox sets up the velocity of the motion, and a reversing device on the output shaft of the feed gearbox sets up the direction

The third kinematic group, producing the tangential feed motion F_{s*} is used to traverse the hob axially However, *the hob cannot be traversed axially without rotation of the gear blank*, since the hob is similar to a rack meshing with a pinion Therefore motion F_{s*} (T_6R_5) is complex and is a formative motion that shapes the tooth profile Hence, the tooth profile is produced by two motions F_v (R_1R_2) and F_{s*} (T_6R_5)

To connect three complex formative motion groups in which three elementary motions are transmitted to the same movable operative member—the gear blank—the hobber requires two differentials This hobber has only one differential however, and the motion groups F_{s2} and F_{s1} are interconnected without any differential in which case the elementary motions R_4 and R_5 are added mathematically and not through a differential Here motion group F_{s2} will be of simple structure and consists of a single internal kinematic constraint in the form of a translatory kinematic pair made up of the tangential cutter head and the hob slide and the drive train The latter must be connected to the internal constraint of the group for motion F_{s1} as has been done in the given hobber Motion F_{s*} is to be set up only in respect to velocity and direction by means of the tangential feed change gears i_{st} *The combination of vertical displacement T_3 and axial displacement T_6* accomplishes the so called oblique feed

The kinematic structure of the hobber may vary to some extent, depending mainly upon which of the three feed screws is to be used

The setting up procedure for this hobber differs only slightly from that employed to set up ordinary gear hobbers The following setup formulas are derived for the case when a helical gear is to be *hobbed with oblique feed*

Cutting motion $F_v(R_1R_2)$

Setting up the indexing change gears i_x
The basic displacements are

$$1 \text{ revolution of the hob} \rightarrow \frac{k}{z} \text{ revolutions of the gear blank}$$

The kinematic balance equation is

$$1 \times \frac{64}{16} \times \frac{29}{29} \times \frac{29}{29} \times \frac{27}{27} \times 1 \times \frac{58}{58} \times \frac{e}{f} \times i_x \times \frac{33}{33} \times \frac{35}{35} \times \frac{1}{96} = \frac{k}{z}$$

The setup formula is

$$i_x = 24 \times \frac{k}{z} \times \frac{f}{e} \qquad (21)$$

The ratio of change gears $\frac{e}{f}$ may be either $\frac{54}{54} = 1$ or $\frac{36}{72} = \frac{1}{2}$. The latter value is used in cutting gears with more than 160 teeth.

The hobber is set up to the required cutting speed by means of the speed gearbox.

Before considering the setting-up procedure for the feed motion F_{s1} (T_3R_4), we shall determine the setup formula for the oblique-feed change gears i_{ob} which will be required further on.

To this end we combine with a kinematic balance equation the vertical feed s_v and the tangential (axial) feed s_t. Thus

$$\frac{s_v}{10} \times \frac{24}{1} \times \frac{42}{35} \times \frac{40}{40} \times i_{ob} \times \frac{50}{50} \times \frac{36}{36} \times \frac{2}{27} \times \frac{60}{48} \times \frac{27}{27} \times \frac{27}{27} \times \frac{5}{48} \times 12 = s_t$$

$$i_{ob} = 3 \times \frac{s_t}{s_v} \qquad (22)$$

Next, we shall set up the feed motion F_{s1} (T_3R_4) and, in particular, the differential change gears i_y.

First we derive the basic displacements of the final members of this train.

In setting up these change gears, it is necessary to take into consideration the fact that the elementary motion R_5, included in motion F_{s2}, is to be provided by feed motion group F_{s1}. Now, this motion will be F_{s1} $(T_3R_4 \pm R_5)$. Hence, in drawing up the basic displacements, motion R_5 must be taken into account. With this exception, the feed train for both motion groups F_{s1} and F_{s2} is one and the same. On the basis of the above-given conditions we can work out the basic displacements for a nondifferential setup, taking into account the velocity parameters of each motion, i.e. taking the vertical

and tangential feeds into account

s_t mm axial travel $\leftarrow s_v$ mm vertical travel of the hob $\rightarrow \frac{s_v}{P}$ revolutions of the gear blank in cutting the helix

$\rightarrow \frac{s_t \cos\lambda}{\pi m_t}$ revolutions of the blank to obtain tangential feed

where λ = helix angle of the hob thread

If the values $P = \frac{\pi m_n}{\sin\beta}$ and $m_t = \frac{m_n}{\sin\beta}$ are substituted in the preceding relationship, then the basic displacements for the differential gear train will be

s_v mm of hob travel $\rightarrow \frac{s_v \sin\beta \pm s_t \cos\lambda}{\pi m_n}$ revolutions of the gear blank

A third component must be added to these basic displacements.

Helical gearing $\frac{16}{64}$ is provided at the hob spindle in the indexing gear train for interconnecting rotation of the hob and of the gear blank. In cutting all types of gears in the hobber, except in three cases (in cutting worm wheels with a hob using tangential feed, in cutting worm wheels with a fly cutter and in cutting helical gears with a hob using oblique feed), these gears operate as ordinary helical gearing. In the three cases mentioned above, when axial travel s_t is imparted to the cutting tool in addition to its rotation (the hob spindle has a thread with pitch $t = 12$ mm for this purpose), the driven gear $64T$ has two degrees of freedom (rotation and axial travel) and this helical gearing becomes a differential. This is due to the fact that gear $64T$ has two rotary motions from two different driving sources: from the rotation of gear $16T$ and from the feed screw $t = 12$ mm.

During its axial motion, the teeth of gear $64T$ slide along the helical teeth of gear $16T$. This imparts additional rotation to gear $64T$. Thus, this helical gearing is a hidden summation transmission, or differential.

In our case, the hidden differential is a part of the internal constraint in the indexing gear train. Hence, the additional rotation of the hob, due to the second driving source of the hidden differential, interferes with the operation of the indexing gear train. Consequently, this additional rotation must be eliminated either by identical rotation in the reverse direction, in which case another differential, not available in the given hobber, must be provided in the indexing train, or by additional rotation transmitted to the gear blank. In the latter case, this additional rotation of the gear blank must be correlated with the additional rotation of the hob so as to provide an additional generating motion. This additional rotation of the gear blank is provided for in setting up the differential gear train.

Upon axial travel of the hob effected by feed screw $t = 12$ mm over the length of the selected tangential feed, the gear $64T$, and therefore the hob

rotate through $\frac{s_t \tan \beta_{64}}{\pi m_f z_{64}}$ revolution; substituting $m_f = \frac{m_n}{\cos \beta_{64}}$, we obtain $\frac{s_t \sin \beta_{64}}{\pi m_n z_{64}}$.

Then, substituting the actual values of $\beta_{64} = 20°20'$ and $m_n = 4$, found in the kinematic diagram, we obtain $\frac{0.34748 s_t}{\pi \times 4 \times 64}$. If this last value is multiplied by the ratio $\frac{k}{z}$, we obtain the additional number of revolutions of the gear blank that can compensate for the operation of the hidden differential. It is equal to $0.00043 \frac{k}{z} s_t$.

Taking into consideration that the oblique-feed change gears provide a quite definite ratio of vertical and tangential feeds, as indicated above, we can substitute the value $s_t = \frac{s_v i_{ob}}{3}$ to obtain the final expression for the basic displacements of the final members of the differential. Thus

$$s_v \text{ mm vertical travel of the hob} \rightarrow$$

$$\rightarrow \frac{s_v \sin \beta}{\pi m_n z} \pm \frac{s_v i_{ob} \cos \lambda}{3 \pi m_n z} \pm \frac{0.00014 i_{ob} k}{z} \text{ revolutions of the gear blank}$$

where s_v = vertical feed of the hob, mm per rev
β = helix angle of the gear being hobbed
m_n = normal module, mm
z = number of teeth on gear being hobbed
i_{ob} = ratio of the oblique-feed change gears
λ = helix angle of the hob thread
k = number of starts on the hob.

The kinematic balance equation for the differential gear train can be drawn up in accordance with these basic displacements:

$$\frac{s_v}{10} \times \frac{24}{1} \times \frac{42}{35} \times i_y \times \frac{27}{27} \times \frac{1}{36} \times 2 \times \frac{58}{58} \times \frac{e}{f} \times i_x \times \frac{33}{33} \times \frac{35}{35} \times \frac{1}{96} =$$

$$= \frac{s_v \sin \beta}{\pi m_n z} \pm \frac{s_v i_{ob} \cos \lambda}{3 \pi m_n z} \pm \frac{0.00014 i_{ob} k}{z}$$

After substituting $i_x = 24 \frac{k}{z} \times \frac{f}{e}$, we obtain the setup formula of the differential train change gears:

$$i_y = \frac{7.95775 \sin \beta}{m_n k} \pm \frac{2.65258 i_{ob} \cos \lambda}{m_n k} \pm 0.00111 i_{ob} \qquad (23)$$

The signs in this formula depend upon the directions of the vertical and axial travel.

To determine the value of i_y, it is necessary to know the value of i_{ob} and, consequently, the ratio $\frac{s_t}{s_v}$. The latter is found from the equation

$$\frac{s_t}{s_v} = \frac{l}{B}$$

where l = length of axial travel of the hob, mm

B = length of vertical travel of the hob, mm.

Having calculated the required ratio of the oblique-feed change gears, the nearest value of i_{ob} is selected from the set of available gears. Change gears i_{ob} have fixed axes, and their ratio has quite definite values:

$$\frac{2}{1},\ 1,\ \frac{1}{2},\ \frac{1}{3},\ \text{and}\ \frac{1}{4}$$

The corresponding one of these values is substituted in the setup formula for change gears i_y. The *selected vertical feed is set up by means* of the feed gearbox.

Radial infeed of the hob is effected by the radial infeed screw RS and the feed gearbox.

The hobber has a separate electric motor M_2 which powers the rapid traverse of the operative members.

In cutting spur or helical gears with a prime number of teeth with the aid of a differential, the feed gear train must be positive. Such a positive constraint cannot be provided by a feed gearbox with friction clutches. Because of this, provision has been made for a feed gear train exclusive of the feed gearbox. To this end, change gears may be installed between the input and output shafts of the feed gearbox. The feed motion can be accomplished through these change gears (shown with dashed lines in the diagram). The hobber also has mechanisms for hand traverse of the stanchion and table, and for the chip conveyer.

Master Gear-Hobbing Machine, Model 543

This machine (Fig. 47) is intended for hobbing high-precision index worm wheels up to 800 mm in diameter and of a module up to 6 mm. The increased accuracy is due to the application of a *correction device* which *transmits* supplementary rotation from special cams through a differential to the work table to obtain more uniform rotation of the gear blank.

In cutting ordinary general purpose worm wheels, the generating operative motion F_r is made up of two elementary motions, i.e. F_r (R_1R_2), in which R_1 is hob rotation and R_2 is gear blank rotation. In the given hobber, the generating motion is made up of four elementary motions F_r $(R_1R_2R_3R_4)$, two of which are not uniform. The kinematic group producing this complex operative motion, consists of the following kinematic constraints.

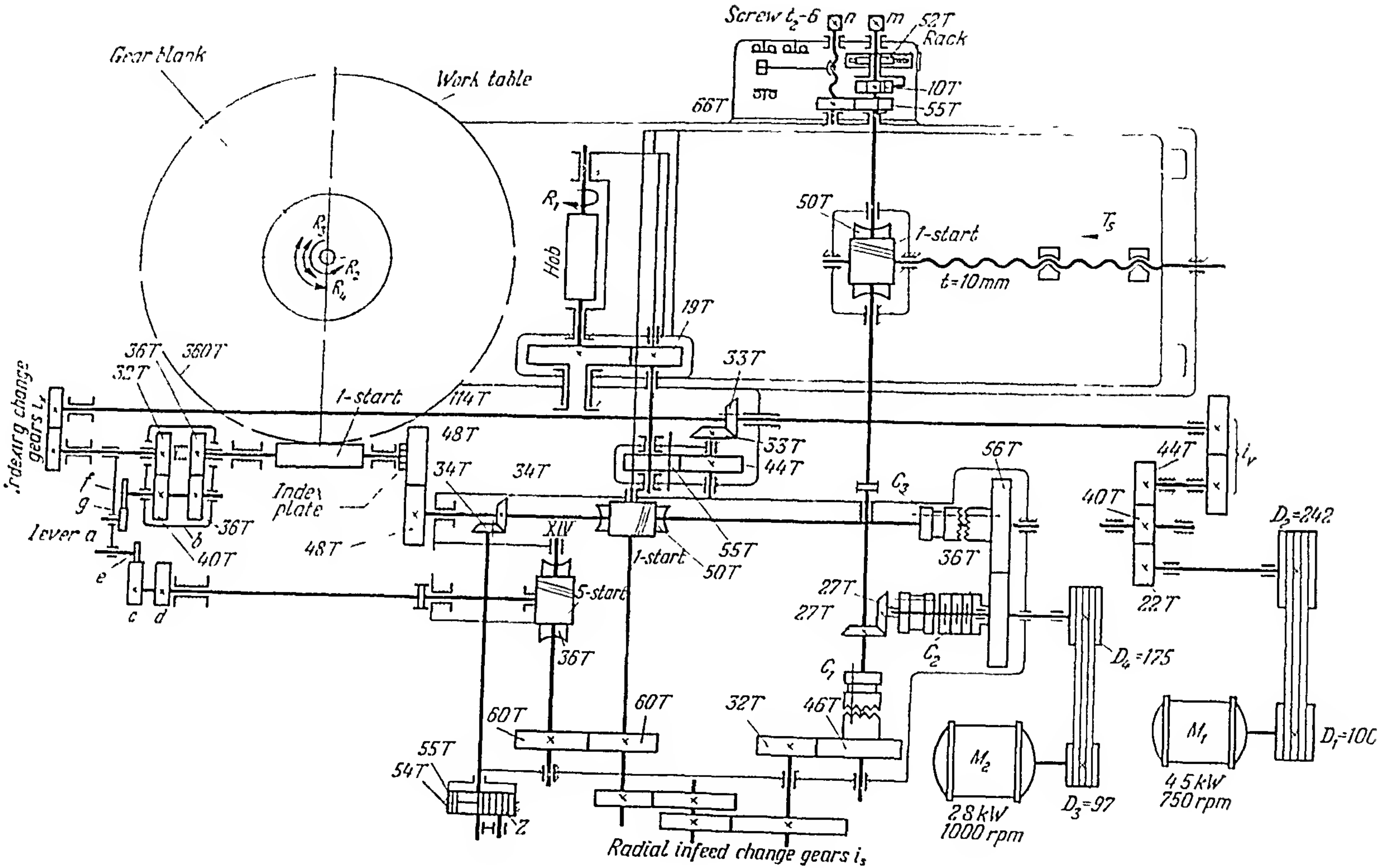

Fig. 47. Kinematic diagram of the master gear-hobbing machine, model 543, made by the ENIMS Stankokonstruktsiya Plant of Moscow (structure class C14)

First gear train of the internal kinematic constraint between uniform hob rotation R_1 and blank rotation R_2:

$$R_1 \to \frac{114}{19} \to \frac{55}{44} \to \frac{33}{33} \to 1_x \to \frac{32}{40} \to$$
$$\to \frac{36}{36} \to \frac{1}{360} \to R_2$$

Second gear train of the same internal constraint between uniform blank rotation R_2 and nonuniform blank rotation R_3 which is actuated by cam c:

$$R_2 \to \frac{360}{1} \to \frac{48}{48} \to \frac{1}{50} \to \frac{60}{60} \to \frac{5}{36} \to$$
$$\to \frac{\Delta r_c}{r_1} \to i_{dif} \to \frac{1}{360} \to R_3$$

where Δr_c = radius increment of cam c (Figs. 47 and 48)

r_1 and r_2 = arm lengths of correction lever a (Fig. 48)

$i_{dif} - \frac{n_{36}}{n_{pc}} = \frac{1}{5}$ = ratio of the differential in transmission from the planet carrier to the sun gear 36T.

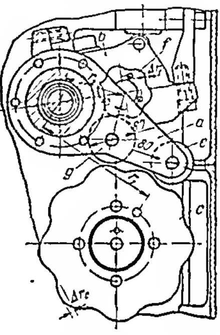

Fig. 48 Correction device of the model 543 gear hobber

Third gear train of the internal constraint between uniform blank rotation R_2 and nonuniform blank rotation R_4 which is actuated by cam f (Figs. 47 and 48):

$$R_2 \to \frac{360}{1} \to \frac{36}{36} \to \frac{\Delta r_f}{r_2} \to i_{dif} \to \frac{1}{360} \to R_4$$

where Δr_f — radius increment of cam f (Figs. 47 and 48)

The correction device, shown in Fig. 48 operates on the following principle. Cam c has a profile corresponding to the accumulated pitch error of the index worm wheel of the hobber and makes one revolution to each revolution of this worm wheel. Rotation of cam c rocks lever a. The latter, through roll g, turns housing b of the differential and, consequently, the work table and gear blank.

In addition to this turning motion, a second cam f, mounted rigidly on the lower shaft of the differential, rotates at the same speed as the index worm of the hobber table. The profile of cam f is laid out on the basis of the cyclic errors of the index worm wheel. Upon the rotation of this cam in respect

to roll g, the differential housing turns and transmits supplementary rotation to the work table. Thus cams c and f, and differential b add the three elementary motions R_2, R_3 and R_4, thereby producing the uniform generating motion F_v $(R_1R_2R_3R_4)$. The curves on the cams are plotted proceeding from a cam rise of 2.1 mm to an angular error of 1″ (sec of arc) in the teeth being cut.

The three kinematic chains indicated above constitute the internal kinematic constraint of the generating motion group. Connected to this constraint is the external constraint:

$$\text{Motor } M_1 \rightarrow \frac{100}{242} \rightarrow \frac{22}{44} \rightarrow i_v \rightarrow \text{gear } 33T$$

The final member, a bevel gear with 33 teeth, interconnects the external and internal constraints. Motion is transmitted from this member to both the hob spindle and the work table. These two constraints constitute the kinematic group which produces the cutting motion (it is also the generating motion).

Besides the generating motion, provision is made for the radial infeed motion FI (T_5). This motion is simple and is produced by a simple kinematic group in which the internal constraint, providing the path of the operative motion, consists of a single translatory kinematic pair formed by the stanchion and the base of the hobber.

The external constraint transmits motion from the table to the feed screw (Fig. 47):

$$R_2 \rightarrow \frac{360}{1} \rightarrow \frac{48}{48} \rightarrow \frac{1}{50} \rightarrow i_s \rightarrow \frac{32}{46} \rightarrow \frac{1}{50} \rightarrow \text{feed screw} \rightarrow T_5$$

A separate electric motor M_2, running at a speed of 1,000 rpm, powers the rapid approach and withdrawal of the stanchion; rapid traverse motion is transmitted through clutch C_2.

The work table can be rotated rapidly to check the radial runout in setting up the blank. This motion is powered by motor M_2 through clutch C_3 with change gears i_x disengaged. Clutch C_1 in the kinematic group for radial infeed disengages the infeed when the preset centre-to-centre distance is reached between the gear blank and the hob.

Besides the continuous radial infeed motion, the hobber has a mechanism for periodic radial infeed which advances the stanchion by an amount ranging from 0.02 to 0.06 mm each time. This mechanism is used in finishing worm wheels. A rack meshing with a pinion $52T$ is shifted hydraulically after each revolution of the work table. The rack pinion turns a pawl and a ratchet wheel which transmits the periodic motion to the feed screw. When the required depth of tooth is reached, the periodic radial infeed is disengaged. This is effected by a nut carrying a stop. This nut mates with and is

traversed by a screw which is driven from the feed screw through the spur gears $55T$ and $66T$

In addition to the correction mechanism the design of the hobber incorporates devices enabling high accuracy to be more dependably maintained in cutting precision worm wheels Thus the helical gear on the hob spindle is of a diameter 3 5 times that of the hob All the sliding joints of the shafts have keys (splines) integral with the shafts The construction of the hob slide and stanchion provides exceptionally high rigidity, and these units are machined to a high degree of accuracy The index worm wheel has 360 teeth of increased depth and with a reduced pressure angle These features reduce the cyclic error of the worm wheel

Dial indicators are built into the index worm housing and table guard to check the runout of test bands on the table and index worm This arrangement checks the operation of the index worm gearing at any time An index plate is mounted on the worm shaft it is used in conjunction with a dial indicator device to check the accuracy of table rotation

Braking devices provided on the shank of the table and on the hob spindle ensure that backlash is eliminated in one direction in the indexing gear train Most of the intermediate shafts run in antifriction bearings

Excess backlash in the worm drive of the correction mechanism is eliminated by axial adjustment of the worm which has a thread varying in thickness from one end of the worm to the other This is known as a dual lead worm

The hobber has an automatic centralized lubricating system During operation a hydraulic device relieves the load on the table ways The electric motors are arranged outside the base Worm wheels of 3rd class accuracy (USSR Std GOST 3675 56) can be cut in this hobber

The hobber is set up by means of three sets of change gears i_x i_v and i_s for which the following basic displacements kinematic balance equations and setup formulas have been derived

1 *Indexing change gears* i_x

The basic displacements are

$$1 \text{ revolution of the hob} \longrightarrow \frac{k}{z} \text{ revolution of the gear blank}$$

The kinematic balance equation is

$$1\times\frac{114}{19}\times\frac{55}{55}\times\frac{33}{33}\times i_x \times \frac{3}{0}\times\frac{3}{3}\times\frac{1}{360}=\frac{k}{z}$$

The setup formula is

$$i_x = 60\frac{k}{z} \qquad (24)$$

2. *Cutting speed change gears* i_v

The basic displacements are

$$750 \text{ rpm of motor } M_1 \longrightarrow n_h \text{ rpm of the hob}$$

The kinematic balance equation is

$$750 \times \frac{100}{242} \times \frac{22}{44} \times i_v \times \frac{33}{33} \times \frac{44}{55} \times \frac{19}{114} = n_h$$

The setup formula is

$$i_v = \frac{n_h}{20} \tag{25}$$

3. *Radial infeed change gears* i_s

The basic displacements are

$$1 \text{ revolution of the gear blank} \longrightarrow s_r \text{ mm of stanchion travel (infeed)}$$

The kinematic balance equation is

$$1 \times \frac{360}{1} \times \frac{48}{48} \times \frac{1}{50} \times i_s \times \frac{32}{46} \times \frac{1}{50} \times 10 = s_r$$

The setup formula is

$$i_s = s_r \tag{26}$$

Correction devices with mechanical members may vary in construction but they all operate on the same principle by which the kinematic errors of the gear train are represented by the profiles of one or several cams. These cams transmit supplementary motion to the table.

Since all of these correction devices are subject to inertia and introduce corrections for the errors without taking the wear of the machine tool into account, they cannot provide sufficiently accurate operation in certain cases. This has led to the application in recent years of correction devices with various nonmechanical linkages, for example, such as those employed in the master gear-hobbing machine, model PH-30, manufactured by the David Brown Corp. of Great Britain.

Gear-Hobbing Machine, Model PH-30 (David Brown Corp., G. B.)

This hobber (Fig. 49) is designed for cutting precision spur gears up to 760 mm in diameter with a hob. In consequence, its structure has been complicated by the provision of a special correction device with electromechanical members. The generating motion F_v (R_1R_2), usually a two-element motion in hobbers, has become a three-element motion F_v $(R_1R_2R_3)$ in this machine. It is made up of hob rotation R_1, uniform table rotation R_2 and nonuniform corrective table rotation R_3.

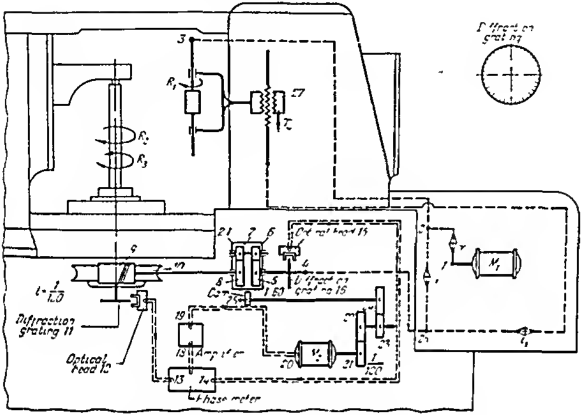

Fig. 49. Structural diagram of the master gear hobbing machine, model PH-30, made by the David Brown Corp., G. B. (structure class K21)

In this kinematic group the internal kinematic constraint consists of two internal kinematic chains. The first internal train interconnects hob rotation R_1 with table rotation R_2, and consists of the following sections of a chain with mechanical members: $R_1 \to 3 \to 2 \to 26 \to 4 \to$ spur gears in the differential $\frac{5}{6} > \frac{7}{8} \to$ worm gearing $\frac{9}{10} \to R_2$. The second internal kinematic chain interconnects uniform rotation R_2 with nonuniform corrective motion R_3 of the table, and consists of branches having both mechanical members and electromechanical devices.

The correction device operates as follows. A precise diffraction grating *11* is mounted on the work table. A light beam from an illuminator passes through the grating to an optical head *12* in which the corresponding electric pulses are produced. The latter are transmitted along section *12-13* to phase

meter *17*. Other pulses, produced by another diffraction grating *16* and optical head *15* mounted at the right-hand side of the differential, are also fed into the phase meter through input *14*. After comparing these pulses, the phase meter issues a signal which is amplified (amplifier *18-19*) and supplied to motor M_2. The latter turns gears *21*, *22*, *23* and *24*, and cam *25* which turns the housing of the differential and, consequently, the work table to correct motion R_2. The drive of these two internal kinematic chains is the mechanical train *1-2* from motor M_1.

The second formative motion, the feed motion F_s (T_4), is of the simple rectilinear type. The kinematic group of this motion is simple and requires no further explanation. The hobber is set up to the required rate of feed by means of the feed change gears i_s as in other hobbers.

5-4. Kinematic Structure of Gear-Cutting Machines Using a Rotary Gear-Shaper Cutter

Methods of Generating Teeth with a Gear-Shaper Cutter

Rotary gear-shaper cutters are employed in gear cutting with various relative positions of the cutter and gear blank axes. There are three possible cases.

If the axes of rotation of the cutter and blank are parallel to each other (Fig. 50*a*), then the cutter can be used to generate spur and helical gears. All gear shapers operate on this principle. In cutting a helical gear, these shapers perform two formative motions: the helical cutting motion F_v (T_3R_4), producing an auxiliary helical gear, and the feed motion, which is also the generating motion F_s (R_1R_2). The shapes of the gear-shaper cutter and the cut gear, and the location of their axes, as well as the motions of the gear shaper, correspond to ordinary helical gearing in which one of the gears has an additional rapid helical reciprocating motion.

The axes of the cutter and blank are in the same position in cutting a spur gear, the only difference being that the cutter performs rectilinear motion F_v (T_3) instead of the helical motion F_v (R_4T_3).

If the axes of rotation of the gear blank and gear-shaper cutter are perpendicular to each other (Fig. 50*b*), the cutter can generate a multiple-start worm using the two motions F_v (R_1R_2) and F_s (T_3R_4). The first of these, the cutting motion F_v (R_1R_2), must be of the same type as the motion in worm gearing in which the cutter plays the part of the worm wheel, while the feed motion F_s (T_3R_4) must be of the same type as in a rack-and-pinion drive in which the cutter represents the pinion. The arrangement of the axes conforms to that of the axes of the elements of worm gearing.

The inclined position of the gear-shaper cutter and blank axes (Fig. 50*c*) corresponds to the position of helical gearing on crossed axes. It is known

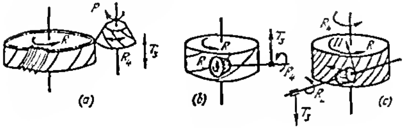

Fig. 50. Motion diagrams for various relative positions of the axes of cutter and gear blank rotation

that upon the rotation of such gears the tooth profile of one gear slides along the tooth of the mating gear. The relative sliding motions of the profiles are utilized here as the cutting motion. Hence, in this case, the generating motion is accomplished at the velocity of the cutting speed F_v (R_1R_2) while the cutting edges travel helically along the tooth with the feed motion F_s (T_3R_4). This differs from ordinary gear shapers in which the generating motion F_v (R_1R_2) was the slow feed motion F_s (R_1R_2). Hence the usual gear shaping method has been converted into a *turning method*. Gear cutting machines producing spur and helical gears with a gear shaper cutter by this turning and generating method in which the cutter and blank axes are crossed have a rapid generating motion F_v (R_1R_2) and a slow motion of the cutter along the tooth—*the feed motion* F_s (T_3R_4).

Both spur and helical gears can be cut in such machines. In cutting a spur gear, a helical gear shaper cutter is required so that the gear can be cut with crossed axes of the cutter and blank. Helical gears are also cut with a helical gear shaper cutter, but a spur cutter could be used as well.

Vertical Gear Shaper, Model 5B150

This gear shaper (Fig. 51) is intended for cutting internal and external spur and helical gears up to 800 mm in diameter and with a module up to 12 mm.

We shall consider the structure and setting up procedure for the case in which an external helical gear is to be cut.

To do this the shaper must have
helical motion F_v (T_3R_4) and the
to these two motions the shaper
approach and withdrawal of the work table, retraction and advance of the cutter to the blank for each stroke, and rapid rotation of the table required to check the radial runout of the gear blank.

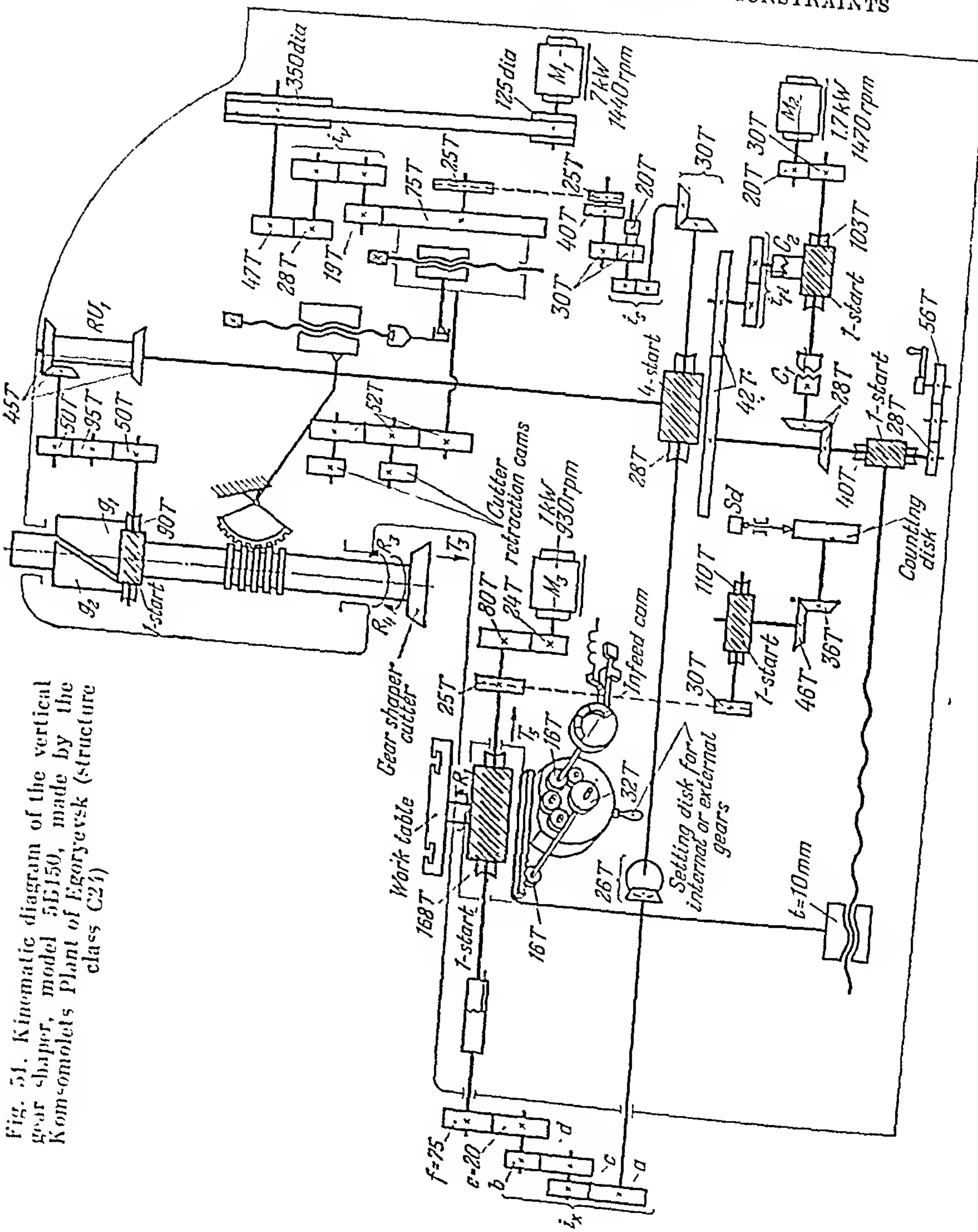

Fig. 51. Kinematic diagram of the vertical gear shaper, model 5B150, made by the Komsomolets Plant of Egoryevsk (structure class C21)

The structure of the cutting motion group F_c $(T_3 R_4)$ consists of an internal constraint in the form of a helical kinematic pair (interchangeable helical guide g_1 *on the worm wheel and interchangeable helical guide* g_2 *on the* cutter spindle or ram) and an external constraint transmitting motion from motor M_1 to the internal constraint.

The cutting motion—a complex motion with an open path—is set up in respect to all five parameters: the interchangeable helical guides set up the path, the crank radius is varied to set up the path length, change gears i_v set up the velocity, the position of the crankpin is changed to set up the direction of the motion, and the length of the connecting rod is changed to set up the initial position.

The kinematic group of the feed motion F_s $(R_1 R_2)$ consists of an internal train between the cutter and gear blank and an external constraint between the motor and worm gearing $\frac{4}{28}$. The worm of the latter is the point connecting the external and internal constraints. The generating motion is complex but has a closed path and is therefore to be set up in respect to three parameters: the indexing change gears i_x and reversing unit RU_1 set up the path of the motion; change gears i_s set up the velocity (rate of feed); they also set up the direction of motion (by means of an idler gear).

The gear shaper is to be set up in accordance with the formulas derived below. First, we shall determine the pitch P_g of the helix on the interchangeable guides g_1 and g_2. In the following, P_{gb} and P_c will denote the pitch (or rather lead) of the helical teeth on the gear blank and cutter, respectively, and z_{gb} and z_c denote the numbers of their teeth.

The following basic displacements of the final members are required to *obtain helical teeth on the gear blank*:

1 revolution of the gear blank $\rightarrow P_{gb}$ mm longitudinal travel of the cutter.

In this, as in other gear shapers, cutter travel over a distance P_{gb} is accompanied by rotation of the cutter which should make $\frac{z_{gb}}{z_c}$ revolutions to each revolution of the gear blank. Hence during $\frac{z_{gb}}{z_c}$ revolutions of the cutter the ram travels P_{gb} mm but since $P_{gb} = \frac{z_{gb}}{z_c} P_c$ then $\frac{z_{gb}}{z_c}$ revolutions of the cutter $\rightarrow \frac{z_{gb}}{z_c} P_c$ mm. It follows that 1 revolution of the cutter $\rightarrow P_c$ mm and $P_g = P_c$.

The ratio i_x of the indexing change gears is determined from the following kinematic balance equation:

$$1 \text{ revolution of the cutter} \quad \frac{90}{1} \times \frac{41}{30} \times \dots \times \frac{28}{1} \times \frac{4}{28} \times i_x \times \frac{e}{f} \times \frac{1}{118} = \frac{z_c}{z_{gb}}$$

thus

$$i_x = \frac{4}{15} \times \frac{f}{e} \times \frac{z_c}{z_{gb}} \tag{27}$$

The purpose of the change gears $\frac{e}{f}$ with fixed axes is to extend the range of the number of teeth cut on gears.

Next we can write the basic displacements and kinematic balance equation for determining the ratio i_s of the feed change gears:

1 revolution of the cutter $\rightarrow K$ full strokes (up and down) of the ram

$$1 \times \frac{90}{1} \times \frac{50}{50} \times \frac{45}{45} \times \frac{28}{4} \times \frac{30}{30} \times \frac{1}{i_s} \times \frac{25}{25} = K$$

$$i_s = \frac{630}{K} \tag{28}$$

The value K is selected in accordance with the required cutting speed.

The radial infeed group is powered by a separate motor M_2.

The ratio i_{fi} of the infeed change gears is determined from the equation

$$1{,}470 \times \frac{20}{30} \times \frac{1}{103} \times i_{fi} \times \frac{42}{42} \times \frac{1}{40} \times 10 = s_r$$

$$i_{fi} = \frac{103}{245} s_r \tag{29}$$

Motor M_2 can transmit rapid traverse motion to the table through clutch C_1. Clutches C_1 and C_2 are interlocked.

Rapid rotation can be transmitted to the table from motor M_3 when index change gears i_x are disengaged. After the finish cutting of the gear is completed and the table has made a preset number of revolutions, a counting disk switches off the gear shaper.

Spur and Helical Gear Generator, Model E3-13

This machine cuts spur and helical gears of a diameter up to a module up to 5 mm by a method resembling thread generation gear-shaper cutter, the axes of the cutter and gear blank bein a definite angle. Two complex formative operative motions F_v F_s (T_3R_4) are produced in the generator (Fig. 52*a*).

The kinematic structure of the generator consists of two compl ic groups (Fig. 52*b*). The first group, producing the cutting sists of an internal kinematic chain with change gears i_x (i train), interconnecting cutter rotation R_1 with gear blank and a drive train from electric motor M_2 through the cuttin

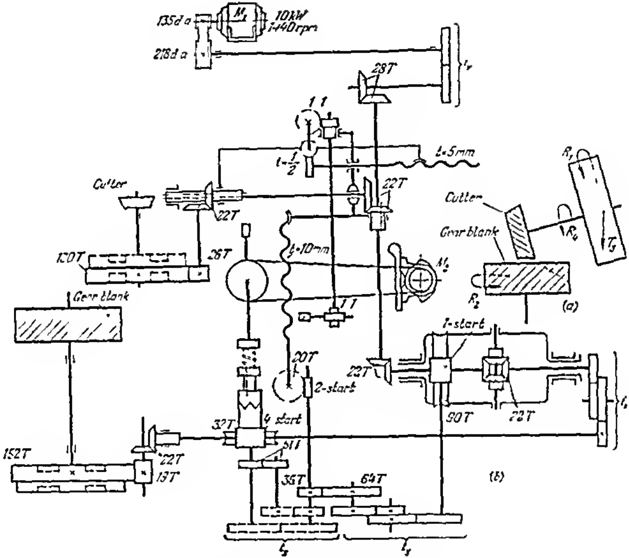

Fig. 52 Kinematic diagram of the spur and helical gear generator, model 5311, made by the Komsomolets Plant of Egoryevsk (structure class C23)

gears i_v to the upper bevel gearing $\frac{22}{22}$. This group produces the tooth profile, provides for the indexing process and is partly responsible for shaping the tooth along its length. Since the internal indexing gear train operates at high speeds, the worm gearing usually employed on the ram and work spindle of gear shapers has been replaced in this generator by spur gearing. F_v is a complex closed motion and is therefore to be set up in respect to three parameters: change gears i_x set up the path, while change gears i_v set up the velocity and direction.

The second kinematic group—the feed motion F_s (T_3R_4)—consists of an internal kinematic chain with change gears i_y (called the differential gear train), interconnecting cutter travel along an element of the gear blank and blank rotation, and a drive train from the motor through change gears i_v, sun gears of the differential, change gears i_x, worm gearing $\frac{4}{32}$ and change gears i_s. Motion is transmitted further through the internal train to the vertical feed screw, and through change gears i_y, planet carrier (housing) of the differential and change gears i_x to the work spindle. Motion F_s is complex and has an open path. Hence, it is to be set up in respect to all five parameters: change gears i_y set up the path; change gears i_s set up the velocity and direction; and dogs operating limit switches that switch the corresponding motors on and off are set up to the path length and initial point of this motion. The cutter slide is returned to its initial position by motor M_2.

The generator is set up in accordance with the formulas derived below.

Cutting motion $F_v(R_1R_2)$

The kinematic balance equation for the indexing gear train is

$$1 \text{ revolution of the cutter} \times \frac{130}{26} \times \frac{22}{22} \times \frac{22}{22} \times \frac{22}{22} \times 1 \times i_x \times \frac{22}{22} \times \frac{19}{152} = \frac{z_c}{z_{gb}}$$

therefore

$$i_x = \frac{8}{5} \times \frac{z_c}{z_{gb}} \tag{30}$$

Setting up the generator to the cutting speed

$$1{,}440 \text{ (rpm of the electric motor)} \times \frac{135}{218} \times i_v \times \frac{28}{28} \times \frac{22}{22} \times \frac{22}{22} \times \frac{26}{130} = n_c$$

whence

$$i_v \cong \frac{n_c}{178} \tag{31}$$

Feed motion $F_s(T_3R_4)$

Setting up the path of the motion

The kinematic balance equation for the differential gear train is

$$1 \text{ revolution of the gear blank} \times \frac{152}{19} \times \frac{22}{22} \times \frac{1}{i_x} \times \frac{1}{2} \times \frac{90}{1} \times \frac{1}{i_y} \times \frac{64}{36} \times \frac{2}{20} \times 10 =$$
$$= P \text{ mm of longitudinal cutter travel}$$

where $P = \frac{\pi m_n z_{gb}}{\sin\beta}$.

After substituting the value for i_x determined above we obtain

$$i_y = \frac{127.3296 \sin\beta}{m_n z_c} \tag{32}$$

Setting up the velocity of the motion

$$1 \text{ revolution of the gear blank} \times \frac{152}{19} \times \frac{22}{22} \times \frac{4}{32} \times i_s \times \frac{2}{20} \times 10 =$$
$$= s_v \text{ mm vertical travel of the cutter}$$

Therefore

$$i_s = s_v \tag{33}$$

Since, in this machine, the cutting motion $F_v\ (R_1R_2)$ is made up of two rotary motions and is continuous, the production capacity is very high in comparison with gear shapers having a reciprocating cutting motion. On the other hand, due to the large number of cutting edges on the cutting tool when the cutter and blank axes are crossed, the tooth shaping process does not proceed properly and the accuracy of cutting in this generator is considerably lower than in a gear shaper. For this reason, model E3-13 is employed, as a rule, for roughing spur and helical gears.

Semiautomatic Worm-Thread Generator, Model E3-10A

This machine is intended for generating the thread of single- or multiple-start worms up to 100 mm in diameter with a maximum pitch of 15 mm and maximum length of 300 mm, using a rotary tool similar to a gear-shaper cutter (Fig. 53a).

The structure of the thread generator (Fig. 53b) consists of two complex *kinematic groups for producing two formative motions: the cutting motion* $F_v\ (R_1R_2)$ and the helical feed motion $F_s\ (T_3R_4)$.

The internal train of the cutting motion group passes from the cutter spindle through change gears i_0, sun gears of the differential, and change gears i_x to the work spindle. The drive train transmits motion from motor M_1 through V-belts, change gears i_v and gearing to bevel gears $\frac{z_c}{z_c}$ which interconnect the drive train and the internal train.

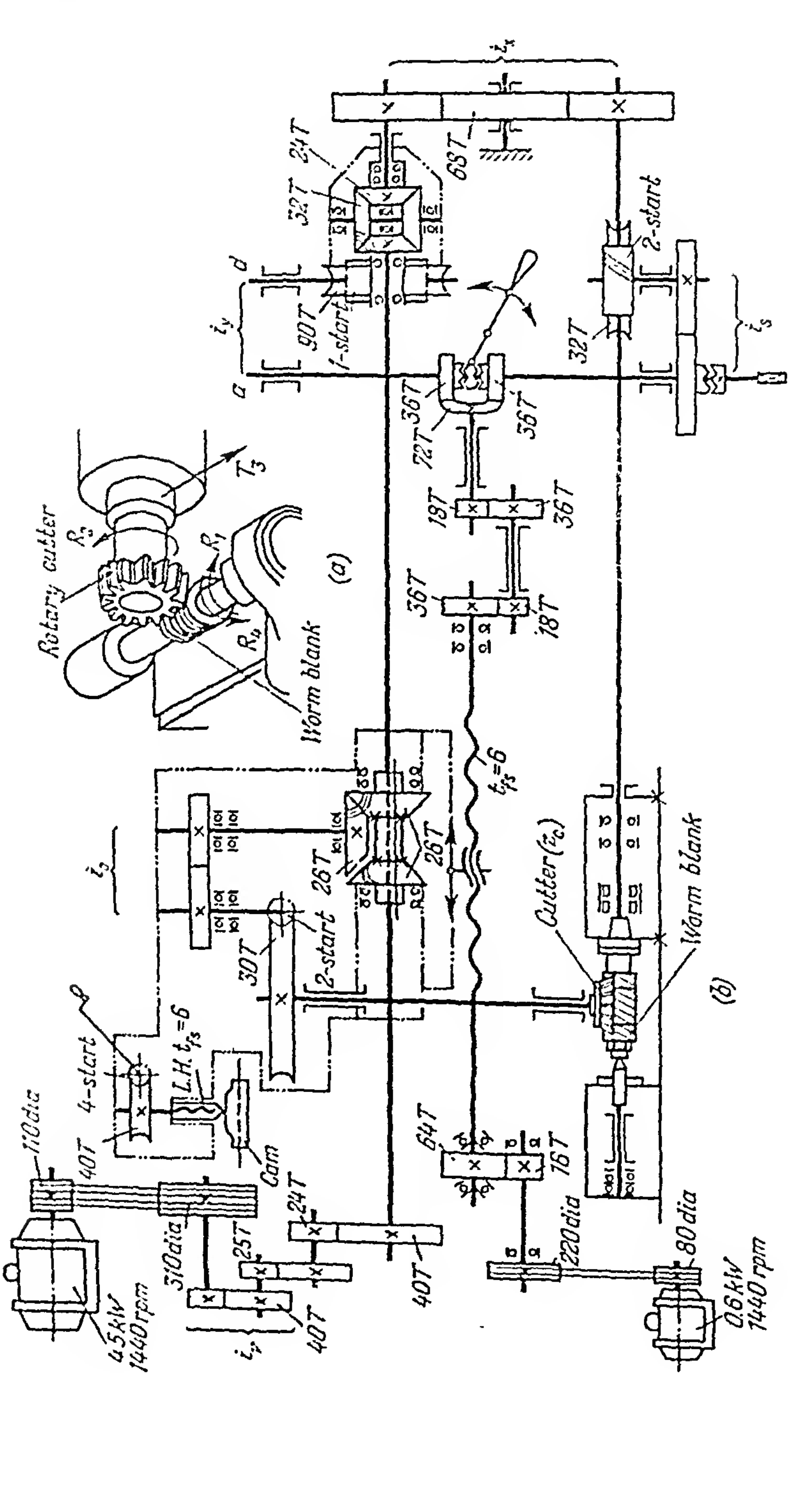

Fig. 53. Kinematic diagram of the semiautomatic worm-thread generator, model E3-10A, made by the Komsomolets Plant of Egoryevsk (structure class C24)

The internal train of the feed group connects the feed screw with the worm blank through change gears i_y worm gearing $\frac{1}{30}$, differential housing and change gears i_x The drive train to the internal train transmits motion from motor M_1 through V belts change gears i_r spur gearing change gears i_x worm gearing $\frac{2}{3\_}$ and feed change gears i_s to the reversing device with bevel gears $36T$ and $72T$ by means of which the drive train and internal train are connected

The generator has two reversing devices They are arranged in the internal trains of the cutting and feed motion groups and enable both right and left hand worms to be generated

Rapid traverse of the slide is effected when the feed screw is powered from a separate electric motor $M_\_$ which can be switched on only after reversing device $\frac{36}{72}$ is set to the neutral position

The upper part of the slide can be swivelled to set the cutter to the required depth of cut and to retract it at the end of the longitudinal stroke The cutter is automatically retracted from the work by a cam

The generator is set up in accordance with setup formulas derived on the basis of the following basic displacements and kinematic balance equation

To determine the ratios of change gears i_x and i_0

1 revolution of the rotary cutter $\rightarrow \frac{\ }{k}$ revolutions of the worm blank

$$1 \times \frac{30}{\_} \times \frac{1}{i_0} \times \frac{26}{\_6} \times 1 \quad i_x = \frac{\_}{k}$$

and

$$i_x = i_0 \frac{\ }{15k} \qquad (35)$$

where z_c = number of teeth on the rotary cutter

k — number of starts on the worm to be generated

Here two sets of change gears i_0 and i_x are located in a single gear train Change gear unit i_0 has fixed axes and only a small number of change gears are available A large set of change gears is available for the change gear unit i_x but it is insufficient for all the possible values of the ratio i_x This is due in part to the fact that, at the large loads transmitted through change gears i_x the latter cannot be designed with an adjustable stud Instead of the latter a series of threaded holes are provided in the base wall in the space between the driving and driven shafts of the change gears An intermediate stud is screwed into one of these holes This change gear construction enables heavy loads to be transmitted and permits a large number of different setup The setup are insufficient however to handle all possible

cases and, because of this, change gears i_0 are provided in the indexing train.

The ratio i_v of the speed change gears can be determined from the equation

$$1{,}440 \times \frac{110}{310} \times i_v \times \frac{25}{40} \times \frac{24}{40} \times 1 \times i_x = n_{wb} \text{ rpm of the worm blank}$$

then

$$i_v = \frac{kn_{wb}}{12 i_0 z_c} \tag{35}$$

The ratio i_y of the differential change gears can be determined by either of two equivalent methods.

First method:

1 revolution of the worm blank $\longrightarrow P$ mm of longitudinal travel of the cutter where P is the lead of the worm thread, mm.

Then

$$1 \times \frac{1}{i_x} \times \frac{1}{2} \times \frac{90}{1} \times \frac{1}{i_y} \times \frac{36}{72} \times \frac{18}{36} \times \frac{18}{36} \times 6 = P$$

and

$$i_y = \frac{2{,}025}{4\pi m i_0 z_c} \tag{36}$$

Second method:

1 revolution of the cutter $\longrightarrow \pi m z_c$ mm of longitudinal travel of the cutter

Then

$$1 \times \frac{30}{2} \times \frac{1}{i_0} \times \frac{26}{26} \times \frac{1}{2} \times \frac{90}{1} \times \frac{1}{i_y} \times \frac{36}{72} \times \frac{18}{36} \times \frac{18}{36} \times 6 = \pi m z_c$$

and

$$i_y = \frac{2{,}025}{4\pi m i_0 z_c} \tag{37}$$

The ratio i_s of the feed change gears is determined as

1 revolution of the worm blank $\longrightarrow s$ mm of longitudinal travel of the cutter

Then

$$1 \times \frac{2}{32} \times i_s \times \frac{36}{72} \times \frac{18}{36} \times \frac{18}{36} \times 6 = s$$

Thus

$$i_s = \frac{64}{3} s \tag{38}$$

Because of the complex cutting tool required, the model E3-10A worm-thread generators find application mainly in large-lot and mass production.

5-5 Kinematic Structure of Spur and Helical Gear Grinders

The sides of gear teeth are finished by the same methods used in cutting the gears but with other cutting tools employed.

The basic type of gear finishing machines are the gear grinders. Most extensively employed are gear grinders using a disk type formed wheel and operating by the forming method, the profile of the wheel being the generating contour, and grinders using a disk type wheel with straight sides and operating by the generating method. In 1940, Vasilchuk, a Soviet engineer, proposed a new high production method of grinding spur and helical gears with a helically profiled wheel, similar to the method of cutting gears with a hob. Insofar as their kinematic structure is concerned, all gear grinders differ only slightly from the gear cutting machines operating on the same principle. The difference in construction is such that gears of higher accuracy are produced. Gear grinders also have an additional formative group for truing and dressing the grinding wheel.

Gear-Grinding Machine, Model 584M

This gear grinder (Fig. 54) is intended for finishing spur and helical gears from 60 to 500 mm in diameter with a module from 2 to 10 mm. It uses a disk type wheel, 260 mm in diameter, with a straight sided cutting profile, representing a tooth of the basic rack.

Spur gears are ground in the following manner.

The rapidly rotating wheel travels rectilinearly along the tooth being ground. At the same time, the work gear rolls in reference to the wheel (in the same way as a pinion rolls along a rack) until the tooth being ground rolls out of mesh with the wheel. After this, the stanchion is withdrawn to the side, the wheel is retracted from the tooth space, and the table with the rotating work gear returns to its initial position. Thus, the involute profile of the tooth is produced by the generating method, and the shape of the tooth along its length is produced by the tangent method. Consequently, three formative motions are required to grind the side surfaces of the teeth: cutting motion F_v (R_1), longitudinal feed motion F_{s1} (T_2) and the roll motion F_{s2} (R_3T_4). Additionally required are the indexing motion Ind (R_5) and the radial infeed motion FI_1 (T_6). Hence, the main part of the grinder consists of five kinematic groups besides the handling motions.

The cutting motion group F_v (R_1) is of the simple type. Its internal constraint consists of a rotary kinematic pair made up of the wheel spindle and the ram. The external constraint consists of a single belt drive between the wheel and motor M_1, also mounted on the ram. The grinding wheel rotates at constant speed (2 200 rpm).

cases and, because of this, change gears i_0 are provided in the indexing train.

The ratio i_v of the speed change gears can be determined from the equation

$$1.440 \times \frac{110}{310} \times i_v \times \frac{25}{40} \times \frac{24}{40} \times 1 \times i_x = n_{wb} \text{ rpm of the worm blank}$$

then

$$i_v = \frac{kn_{wb}}{12 i_0 z_c} \tag{35}$$

The ratio i_y of the differential change gears can be determined by either of two equivalent methods.

First method:

1 revolution of the worm blank $\rightarrow P$ mm of longitudinal travel of the cutter

where P is the lead of the worm thread, mm.

Then

$$1 \times \frac{1}{i_x} \times \frac{1}{2} \times \frac{90}{1} \times \frac{1}{i_y} \times \frac{36}{72} \times \frac{18}{36} \times \frac{18}{36} \times 6 = P$$

and

$$i_y = \frac{2{,}025}{4\pi m i_0 z_c} \tag{36}$$

Second method:

1 revolution of the cutter $\rightarrow \pi m z_c$ mm of longitudinal travel of the cutter

Then

$$1 \times \frac{30}{2} \times \frac{1}{i_0} \times \frac{26}{26} \times \frac{1}{2} \times \frac{90}{1} \times \frac{1}{i_y} \times \frac{36}{72} \times \frac{18}{36} \times \frac{18}{36} \times 6 = \pi m z_c$$

and

$$i_y = \frac{2{,}025}{4\pi m i_0 z_c} \tag{37}$$

The ratio i_s of the feed change gears is determined as

1 revolution of the worm blank $\rightarrow s$ mm of longitudinal travel of the cutter

Then

$$1 \times \frac{2}{32} \times i_s \times \frac{36}{72} \times \frac{18}{36} \times \frac{18}{36} \times 6 = s$$

Thus

$$i_s = \frac{64}{3} s \tag{38}$$

Because of the complex cutting tool required, the model E3-10A worm-thread generators find application mainly in large-lot and mass production.

5-5 Kinematic Structure of Spur and Helical Gear Grinders

The sides of gear teeth are finished by the same methods used in cutting the gears but with other cutting tools employed.

The basic type of gear finishing machines are the gear grinders. Most extensively employed are gear grinders using a disk type formed wheel and operating by the forming method, the profile of the wheel being the generating contour, and grinders using a disk type wheel with straight sides and operating by the generating method. In 1940 Vasilchuk, a Soviet engineer, proposed a new high production method of grinding spur and helical gears with a helically profiled wheel, similar to the method of cutting gears with a hob. Insofar as their kinematic structure is concerned, all gear grinders differ only slightly from the gear cutting machines operating on the same principle. The difference in construction is such that gears of higher accuracy are produced. Gear grinders also have an additional formative group for truing and dressing the grinding wheel.

Gear Grinding Machine, Model 584M

This gear grinder (Fig. 54) is intended for finishing spur and helical gears from 60 to 500 mm in diameter with a module from 2 to 10 mm. It uses a disk-type wheel, 260 mm in diameter, with a straight-sided cutting profile representing a tooth of the basic rack.

Spur gears are ground in the following manner.

The rapidly rotating wheel travels rectilinearly along the tooth being ground. At the same time, the work gear rolls in reference to the wheel (in the same way as a pinion rolls along a rack) until the tooth being ground rolls out of mesh with the wheel. After this, the stanchion is withdrawn to the side, the wheel is retracted from the tooth space, and the table with the rotating work gear returns to its initial position. Thus, the involute profile of the tooth is produced by the generating method and the shape of the tooth along its length is produced by the tangent method. Consequently, three formative motions are required to grind the side surface of the teeth: cutting motion $F_v (B_1)$, longitudinal feed motion $F_{s1} (T_2)$ and the roll motion $F_{s2} (B_3T_4)$. Additionally required are the indexing motion $Ind (B_5)$ and the radial infeed motion $FI_1 (T_6)$. Hence, the main part of the grinder consists of five kinematic groups besides the handling motions.

The cutting motion group $F_v (B_1)$ is of the simple type. Its internal constraint consists of a rotary kinematic pair made up of the wheel spindle and the ram. The external constraint consists of a single belt drive between the wheel and motor M_1, also mounted on the ram. The grinding wheel rotates at constant speed (2,200 rpm).

cases and, because of this, change gears i_0 are provided in the indexing train.

The ratio i_v of the speed change gears can be determined from the equation

$$1{,}440 \times \frac{110}{310} \times i_v \times \frac{25}{40} \times \frac{24}{40} \times 1 \times i_x = n_{wb} \text{ rpm of the worm blank}$$

then

$$i_v = \frac{kn_{wb}}{12 i_0 z_c} \tag{35}$$

The ratio i_y of the differential change gears can be determined by either of two equivalent methods.

First method:

1 revolution of the worm blank $\rightarrow P$ mm of longitudinal travel of the cutter where P is the lead of the worm thread, mm.

Then

$$1 \times \frac{1}{i_x} \times \frac{1}{2} \times \frac{90}{1} \times \frac{1}{i_y} \times \frac{36}{72} \times \frac{18}{36} \times \frac{18}{36} \times 6 = P$$

and

$$i_y = \frac{2{,}025}{4\pi m i_0 z_c} \tag{36}$$

Second method:

1 revolution of the cutter $\rightarrow \pi m z_c$ mm of longitudinal travel of the cutter

Then

$$1 \times \frac{30}{2} \times \frac{1}{i_0} \times \frac{26}{26} \times \frac{1}{2} \times \frac{90}{1} \times \frac{1}{i_y} \times \frac{36}{72} \times \frac{18}{36} \times \frac{18}{36} \times 6 = \pi m z_c$$

and

$$i_y = \frac{2{,}025}{4\pi m i_0 z_c} \tag{37}$$

The ratio i_s of the feed change gears is determined as

1 revolution of the worm blank $\rightarrow s$ mm of longitudinal travel of the cutter

Then

$$1 \times \frac{2}{32} \times i_s \times \frac{36}{72} \times \frac{18}{36} \times \frac{18}{36} \times 6 = s$$

Thus

$$i_s = \frac{64}{3} s \tag{38}$$

Because of the complex cutting tool required, the model E3-10A worm-thread generators find application mainly in large-lot and mass production.

5-5. Kinematic Structure of Spur and Helical Gear Grinders

The sides of gear teeth are finished by the same methods used in cutting the gears, but with other cutting tools employed.

The basic type of gear finishing machines are the gear grinders. Most extensively employed are gear grinders using a disk type formed wheel and operating by the forming method, the profile of the wheel being the generating contour, and grinders using a disk type wheel with straight sides and operating by the generating method. In 1940 Vasilchuk, a Soviet engineer, proposed a new high production method of grinding spur and helical gears with a helically profiled wheel similar to the method of cutting gears with a hob. Insofar as their kinematic structure is concerned, all gear grinders differ only slightly from the gear cutting machines operating on the same principle. The difference in construction is such that gears of higher accuracy are produced. Gear grinders also have an additional formative group for truing and dressing the grinding wheel.

Gear-Grinding Machine, Model 5841t

This gear grinder (Fig. 54) is intended for finishing spur and helical gears from 60 to 500 mm in diameter with a module from 2 to 10 mm. It uses a disk type wheel 260 mm in diameter, with a straight sided cutting profile, representing a tooth of the basic rack.

Spur gears are ground in the following manner.

The rapidly rotating wheel travels rectilinearly along the tooth being ground. At the same time the work gear rolls in reference to the wheel (in the same way as a pinion rolls along a rack) until the tooth being ground rolls out of mesh with the wheel. After this the stanchion is withdrawn to the side, the wheel is retracted from the tooth space and the table with the rotating work gear returns to its initial position. Thus the involute profile of the tooth is produced by the generating method, and the shape of the tooth along its *length is produced by* the tangent *method*. Consequently three formative motions are required to grind the side surfaces of the teeth: cutting motion F_v (R_1), longitudinal feed motion F_{s1} (T_*) and the roll motion F_{s*} (R_3T_i). Additionally required are the indexing motion *Ind* (R_5) and the radial infeed motion FI_1 (T_6). Hence the main part of the grinder consists of five kinematic groups besides the handling motions.

The cutting motion group F_v (R_1) is of the simple type. Its internal constraint consists of a rotary kinematic pair made up of the wheel spindle and the ram. The external constraint consists of a single belt drive between the wheel and motor M_1 also mounted on the ram. The grinding wheel rotates at constant speed (2 200 rpm).

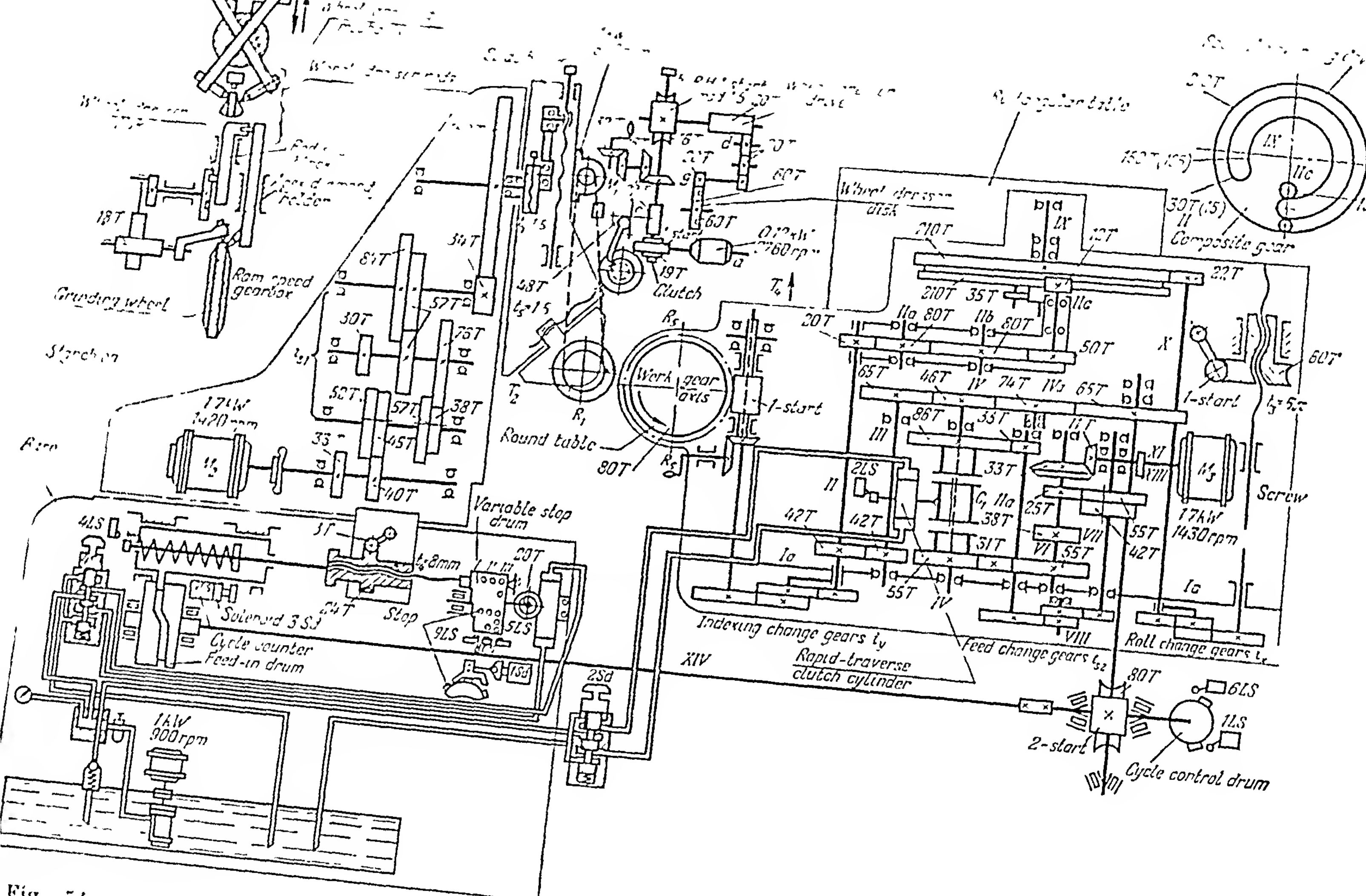

Fig. 54. Kinematic diagram of the gear-grinding machine, model 584M, made by the ENIMS Stankokonstruktsiya Plant of Moscow (structure class K34)

The group of the longitudinal feed motion F_{41} (T_4) is also simple in nature. A translatory kinematic pair in the form of the rectilinear ways on the ram and the stanchion constitutes the internal constraint. The external constraint housed in the stanchion transmits motion from motor M_4 to a crank drive and to the ram. Motion F_{41} (T_4) is a simple motion with an open path and is therefore to be set up in respect to four parameters: eight-step speed gearbox i_{41} sets up the velocity of the motion providing for ram motion in a range from 34 to 268 full strokes (up and down) per minute; the path length is set up by varying the crank radius with a screw having a pitch t_1 = 1.5 mm; the initial point of the motion is set up by changing the position of the ram in reference to the connecting rod with a screw having a pitch t_4 = 5 mm, and the direction is set up by means of the crank disk itself.

The roll motion group F_{44} (B_3T_4) is of the complex type. Its internal constraint consists of a kinematic chain between the round and the rectangular tables interconnecting their elementary motions B_3 and T_4. This train is as follows: motion $B_3 \rightarrow$ worm gearing $\frac{1}{80} \rightarrow$ indexing change gears $i_y \rightarrow$ $\rightarrow$ gears $20T$, $80T$, $80T$ and $50T \rightarrow$ special reversing device consisting of a movable drive pinion $22T$, a composite gear and a driven gear $22T \rightarrow$ roll change gears $i_x \rightarrow$ travel screw with a pitch $5\pi \rightarrow$ motion T_4.

The external constraint of this group transmits motion from a separate motor M_3 to the internal constraint through gear $42T$ mounted on shaft II. This constraint is as follows: $M_3 \rightarrow$ bevel gears $11T$ and $33T \rightarrow$ gears $25T$ and $38T$ and sliding cluster i_0 with gears $55T$ and $42T \rightarrow$ feed change gears i_4, and gears $33T$ and $86T \rightarrow$ clutch C_1 and gears $46T$, $65T$, $42T$ and $42T$. The last spur gear ($42T$) is the junction interconnecting the external and internal constraints. From it motion is transmitted to the round table and simultaneously to the travel screw with a pitch $t_3 = 5\pi$.

Motion F_{44} (B_3T_4) is complex with an open path and is therefore to be set up in respect to all five parameters. It is set up to the path by means of roll change gears i_x, and in respect to velocity by feed change gears i_4. From manufacturing considerations the direction of the roll motion is not changed. The path length of the roll motion is varied by adjustable dogs which operate limit switches GLS and HLS. The required initial position of the rectangular table is set up by hand rotation transmitted through worm gearing and nut to the travel screw with a pitch t_3 = 5π.

The indexing motion group Ind (B_3) is simple and has an internal constraint in the form of a single rotary kinematic pair made up of the round table shaft and the rectangular table.

The external constraint is also powered from motor M_3 of the feed motion group F_{44} and transmits motion to the round table through the lower constraint of the feed motion group F_{42} and partly through the internal constraint of this group.

Motion *Ind* (R_5) is simple and has an open path. Consequently, it should require setting up in respect to four parameters. However, being an indexing motion, it is set up in respect to only one parameter—path length—by means of the indexing change gears i_{ii}. The round table belongs to two groups—indexing and roll—which can be interconnected, as previously pointed out, by three methods. A special reversing device, in the form of a composite gear, is located in the internal constraint of the roll motion group. Hence, in this grinder, a compound method of group interconnection has been applied. This means that in grinding a tooth, the processes of profiling and indexing first proceed simultaneously and then separately.

When the grinding of a tooth is completed, the stanchion with the wheel is withdrawn by the feed-in drum, and a special device reverses the direction of travel of the rectangular table without reversing the rotation of the round table. Therefore, at this moment, the roll motion (R_3T_4) ceases and motion R_3 becomes the indexing motion R_5, while motion T_4 is replaced by the handling motion T_7 which returns the rectangular table to its initial position. During grinding and table return, the work gear turns through an angle corresponding to several teeth z_i. In indexing a work gear to the given number of teeth z_{wg} it is necessary to select z_i so that it and z_{wg} have no factors in common. In the given gear grinder $z_i \gg 7$, and of such a value that the length of the stroke of the rectangular table is sufficient to grind the tooth completely. The value z_i is to be determined from a formula given in the Service Manual of the grinder.

The radial infeed motion group FI_1 (T_6) is simple. Its internal constraint consists of a single translatory kinematic pair formed by the stanchion and the base. Radial traverse of the stanchion is powered by motor M_3 through the following external constraint: $M_3 \rightarrow$ bevel gears $14T$ and $33T \rightarrow$ spur gears $25T$ and $38T$, and cluster gear $i_0 \rightarrow$ feed change gears $i_{s2} \rightarrow$ spur gears $33T$, $86T$, $46T$, $74T$ and $65T \rightarrow$ worm gearing $\frac{2}{80} \rightarrow$ feed-in drum $\rightarrow$ stanchion rod and travel screw with a pitch $t_4 = 8$ mm. This motion is set up only in respect to two parameters: path length and initial point. The path length is set up by the variable stop drum having a number of ground pads of different height on its end face. The initial position of the stanchion is set up by means of the travel screw with a pitch $t_4 = 8$ mm, linked to the stanchion rod and a rotary nut.

The grinder has a number of other kinematic groups as well. For instance, it has a group, found in all grinders, for truing and dressing the grinding wheel. Dressing may be either manual or automatic. This is accomplished by the dressing mechanism drive which provides, not only the dressing motion, but a motion that compensates for wheel wear and is effected by a ratchet and pawl mechanism and a travel screw with a pitch $t_5 = 1.5$ mm. The gear grinder has facilities for three modes of operation: automatic

cycle, semiautomatic cycle and manual controls, the latter being for setting-up purposes.

The grinder operates as follows on an automatic cycle.

After starting the grinder and finishing the grinding of one tooth, a stop on the cycle control drum operates limit switch *1LS* (Fig. 54). This energizes solenoid *2Sd* of the pilot valve controlling clutch C_1 which engages rapid traverse of the table. After the feed-in drum withdraws the stanchion with the grinding wheel from the work gear, the rectangular table, actuated by a special reversing device with a composite gear, begins to return rapidly while the round table continues to rotate in the same direction. One indexing cycle is completed when the rectangular table reaches its initial position and the work gear is rotated further (indexed) through an angle corresponding to z_i teeth. As it is withdrawn, the stanchion operates limit switch *4LS*. If at the end of the return stroke of the rectangular table, limit switch *6LS* is released and the rapid traverse clutch C_1 is shifted to the slow working travel position, then limit switch *2LS* is also released and the grinding cycle of the next tooth begins. If, for some reason, the rapid traverse clutch does not engage working travel, then limit switch *4LS* switches off motor M_2 for longitudinal travel of the ram and motor M_3 for the roll motion.

After all the teeth have been ground in the first pass, the cycle counter, set up to the given number of teeth z_{wg}, operates limit switch *3LS*. At this, solenoid *3Sd* resets the counter to zero and solenoid *1Sd* puts the escapement lever into a position in which the variable stop (control) drum can be turned to a new position by the action of a hydraulic cylinder, rack and rack pinion *20T*.

Three rows of threaded holes, into which stops can be screwed as required, are provided on the periphery of the variable stop drum. These stops operate limit switches *9LS*, *8LS* and *5LS*.

The stops are arranged and installed on the drum in accordance with a predetermined control schedule for the given work gear. If, upon rotation of the drum, a stop in the third row operates limit switch *5LS*, tooth grinding begins again at a new depth of cut on the second pass. If a stop of the second row operates limit switch *8LS*, wheel dressing begins and continues until limit switch *7LS*, located in the dressing drive mechanism, is operated and switches off the dressing mechanism and starts the machine for resuming the grinding of the work gear. Finally, if a stop of the first row operates limit switch *9LS*, the latter transmits a command pulse to end the whole grinding cycle and to return the variable stop drum to its initial position, thereby enabling the next work gear to be ground.

In operation on a semiautomatic cycle, the grinder will automatically stop after grinding all the teeth in one pass at a constant depth of cut. The grinder is set to a new depth of cut and started on the next pass by the operator.

In setting up the machine, the operator uses the manual controls.

A special selector switch is used to change over from one mode of operation to another.

To grind helical gears, the ram is set at the helix angle β of the work gear in relation to the axis of this gear so that the longitudinal motion of the wheel is along the gear tooth. In this position of ram travel, the helix along the tooth length is produced by the tangent method with inclined ram motion F_{s1} and roll motion F_{s2}. Since a helix can be developed on a plane, no co-ordination is required between the longitudinal motion of the ram and work gear rotation in grinding a helical gear. Such a co-ordination is required in cutting a helical gear with a hob if the latter is fed parallel to the gear blank axis. For this reason, gear grinders with a swivelling ram have no differentials.

The gear grinder is set up in accordance with the formulas derived below.

The machine has three change gear units i_y, i_x and i_{s2}. Hence, it will be necessary to draw up three basic kinematic chains.

1. *Indexing gear train with change gears i_y*

The indexing cycle takes place during one revolution of the cycle drum; consequently, the basic displacements are

1 revolution of the cycle drum $\rightarrow \frac{z_i}{z_{wg}}$ revolution of the work gear

Then the kinematic balance equation is

$$1 \times \frac{80}{2} \times \frac{65}{65} \times \frac{42}{42} \times i_y \times \frac{1}{80} = \frac{z_i}{z_{wg}}$$

whence

$$i_y = \frac{2z_i}{z_{wg}} \tag{39}$$

2. *Profiling (roll) gear train with change gears i_x*

The basic displacements are

1 revolution of the work gear $\rightarrow \pi m_i z_{wg}$ mm of rectangular table travel

The kinematic balance equation is

$$1 \times \frac{80}{1} \times \frac{1}{i_y} \times \frac{20}{50} \times \frac{22}{210} \times \frac{210}{22} \times i_x \times 5\pi = \pi m_i z_{wg}$$

where $i_y = \frac{2z_i}{z_{wg}}$.

Then the setup formula is

$$i_x = \frac{m_i z_i}{80} \tag{40}$$

3 Feed gear train with change gears i_{s2}

Gear grinders having a disk type wheel and operating by the generation grinding method use a feed expressed as cycle time It is the time t_c in seconds required to grind one tooth Each tooth is ground during one revolution of the cycle drum Hence, the basic displacements are

$$1 \text{ revolution of the cycle drum} \rightarrow \frac{1,450}{60} t_c \text{ revolutions of the motor}$$

The kinematic balance equation is

$$1 \text{ revolution of the cycle drum} \times \frac{82}{2} \times \frac{16}{36} \quad \frac{86}{34} \times \frac{1}{i_{s2}} \times \frac{1}{i_0} \times \frac{33}{11} = \frac{1,450}{60} t_c$$

The setup formula is

$$i_{s2} \approx \frac{19}{i_0 t_c}$$

Gear-Grinding Machine, Model 5A833

This grinder generates a gear tooth shape with a grinding wheel on which a helical thread has been developed It is capable of grinding spur and helical gears up to 320 mm in diameter and with a module up to 5 mm.

Since the thread profile of the worm-shaped wheel (Fig 55) conforms to the profile of the basic rack teeth in the same way as the thread of a gear hob, the cutting motion F_v, being the formative motion that shapes the gear tooth profile, is of the complex type—$F_v (R_1 R_2)$ This motion is the same as the cutting speed motion in a gear hobber.

The helical shape along the tooth length is obtained in grinding helical gears by inclining the ways along which the workhead with the work gear travels to angle β (β is the helix angle of the gear being ground). The property of the ruled side surfaces of the gear teeth to be developed on a plane, used in the model 584M grinder is also used in this machine If the work gear is slowly traversed with the feed motion $F_s (T_3)$ along the side surface of the tooth at an angle β to an element of the cylinder (Fig 55) then, in conjunction with the rapid roll motion $F_r (R_1 R_2)$, a helical side surface will be ground on the gear

In addition to these formative motions the grinder has two more operative motions (Fig 56) the radial infeed motion $FI (T_4)$ and the complex wheel-dressing (or truing) motion $F_{t\,d} (T_5 R_6)$ in which T_4 is radial infeed of the wheel to the work T_5 is longitudinal travel of the dressing diamond or crusher roll, and R_6 is rotation of the wheel Thus, the main part of the grinder accomplishes four operative motions F_r F_s FI and $F_{t\,d}$ It therefore consists of four separate kinematic groups (handling motions are not to be considered here)

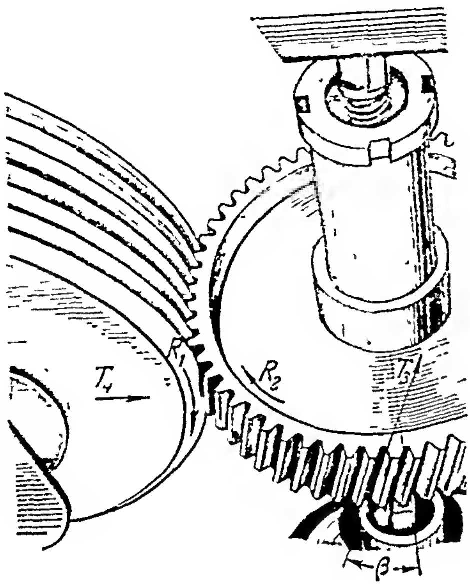

Fig. 55. Operative motions of a gear grinder using a helically profiled wheel (model 5A833)

The internal constraint between the wheel and work spindles in the kinematic group of the cutting motion F_v is of the mechanical type; it comprises four pairs of gears with parallel axes, three pairs of gears with crossed axes and worm gearing with a multiple-start worm. The accuracy with which gears can be ground depends, for the most part, on the operation of this gear train. It is difficult, however, to manufacture such a long train, running at high speed, with sufficient accuracy, and to maintain this accuracy in the course of operation under ordinary shop conditions.

A braking-load pump is linked to the work spindle to make the gear train operate more smoothly. The external constraint of the cutting motion group F_v is very simple; the latter is powered by motor M_1.

The kinematic group of the feed motion F_s (T_3) is simple and is powered from the same motor M_1 through the internal constraint of the first kinematic group and worm gearing $\frac{1}{24}$.

The kinematic group producing motion F_{wd}, for dressing and truing the grinding wheel, consists of an internal constraint between the crusher roll slide and the wheel spindle through change gears i_y. Its external constraint is driven from motor M_2 through speed gearbox i_0.

Hydraulic drive HC_1 is used in the kinematic group for radial infeed. The grinder does not have a differential.

Next we shall derive the setup formulas.

1. *Tooth profiling gear train* (it is also the indexing gear train)

1 revolution of the grinding wheel $\longrightarrow \frac{1}{z_{wg}}$ revolution of the work gear

thus

$$1 \cdot \frac{92}{92} \times \frac{13}{13} \times i_x \times \frac{13}{13} \times \frac{90}{90} \times \frac{13}{13} \times \frac{122}{122} \times \frac{3}{48} = \frac{1}{z_{wg}}$$

and

$$i_x = \frac{16}{z_{wg}} \tag{41}$$

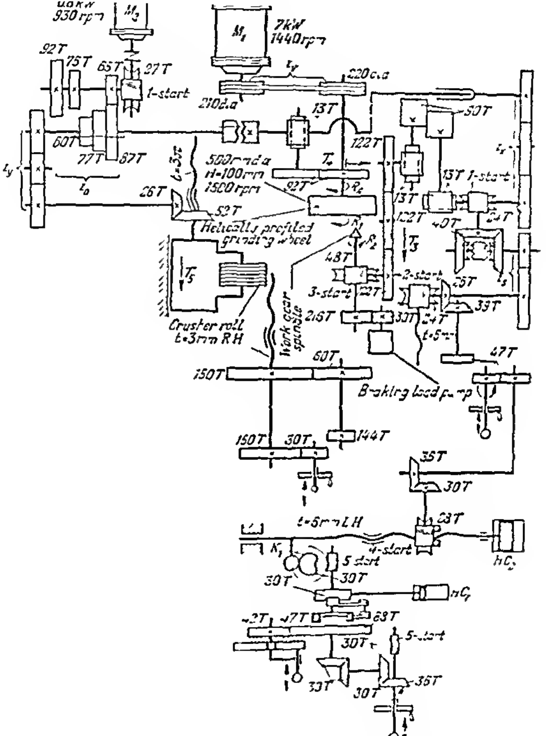

Fig. 56. Kinematic diagram of the gear grinding machine model 5A8?3 made by the Komsomolets Plant at Egoryevsk (structure class Ku...)

2. *Longitudinal feed gear train*

1 revolution of the work gear $\rightarrow s_{lg}$ mm of longitudinal travel of the work gear

thus

$$1 \times \frac{48}{3} \times \frac{122}{122} \times \frac{13}{13} \times \frac{90}{90} \times \frac{13}{13} \times \frac{1}{24} \times \frac{40}{40} \times i_s \times \frac{2}{24} \times 6 = s_{lg}$$

then

$$i_s = 3s_{lg} \tag{42}$$

3. *Gear train for developing a helical thread on the wheel*

1 revolution of the grinding wheel $\rightarrow \tau$ mm of longitudinal travel of the crusher roll

where τ = pitch of the helical thread on the grinding wheel.

Thus

$$1 \times \frac{92}{92} \times \frac{13}{13} \times i_y \times \frac{26}{52} \times 3\pi = \tau = \frac{m_n \pi}{\cos \beta}$$

where β = helix angle of the thread on the grinding wheel
m_n = normal module of the gear being ground, mm.

The setup formula is

$$i_y = \frac{3}{2} \times \frac{m_n}{\cos \beta} \tag{43}$$

Since the angle β is small if the wheel diameter is large (500 mm), $\cos \beta \cong \sim 1$, and the following approximate formula can be employed in setting up the grinder:

$$i_y = \frac{3}{2} m_n \tag{44}$$

4. *The wheel rotation speed gear train for dressing is*

$$930 \times \frac{1}{27} \times i_0 \times \frac{13}{13} \times \frac{92}{92} = n_o$$

and

$$i_0 \cong \frac{n_o}{34} \tag{45}$$

5. *Radial infeed gear train*

Upon each longitudinal stroke of the workhead with the work gear, the work stanchion is automatically infed radially by the action of hydraulic cylinder HC_1 through the ratchet wheel $z_{rw} = 47T$, worm gearing $\frac{5}{30}$ and

radial infeed cam K_1. Hydraulic cylinder HC_2 rapidly retracts the work gear from the wheel. Manual longitudinal and radial traverse of the work gear are also available.

Gears with a module $m < 0.8$ mm are ground from the solid; they require no gear cutting operation.

Semiautomatic Gear-Grinding Machine, Model 588

This machine (Fig. 57) can grind spur gears up to 800 mm in diameter, with a module up to 12 mm by the formed-wheel grinding method.

In grinding a spur gear with a disk type formed wheel, the following motions are required: cutting speed motion F_v (R_1), longitudinal feed motion F_s (T_2), indexing motion *Ind* (R_3) and the vertical infeed motion *FI* (T_4), the latter motion not being shown in the diagram. The last two motions are accomplished by the work gear. In addition to these, handling and controlling motions are also required; they will be considered below.

The main part of the grinder structure, producing the formative, indexing and feed-in motions, consists of four kinematic groups. The group for the cutting speed motion F_v (R_1) is of the simple type. It consists of an internal constraint in the form of a single rotary kinematic pair made up of the wheel spindle and the wheelhead, and an external constraint consisting of a belt drive between motor M_1 and the wheel spindle. There are no setting up devices in this group.

The longitudinal feed motion group F_s (T_2) is also simple. Its internal constraint consists of a single translatory kinematic pair made up of the wheelhead and the base. This group is powered by the hydraulic cylinder HC_1. The motion is to be set up in respect to four parameters (path length, velocity, direction and the initial point) by means of a hydraulic control panel.

The group for the indexing motion *Ind* (R_3) is likewise simple, and its internal constraint consists of a single rotary kinematic pair made up of the work gear spindle and the work head housing. The external constraint consists of indexing change gears i_x and jaw clutch C_1, located between the work spindle and motor M_2. This constraint also includes the train with change gears i_0 between indexing sleeve 5 and time disk 9. The indexing motion *Ind* (R_3) is periodical, and the indexing mechanism operates as follows.

Rotation from motor M_2 is transmitted continuously to driving disk *1* of jaw clutch C_1. Driven disk *2* of this clutch carries a four tooth ratchet wheel *3* which may engage with pawl *4* pivoted in a slot of indexing sleeve *5*. The latter has two index disks *6a* and *6b*, each with a single indexing slot. Two locking members *7* and *8* engage with the slots in the two disks locking them rigidly so that indexing sleeve *5* is held stationary. During this time

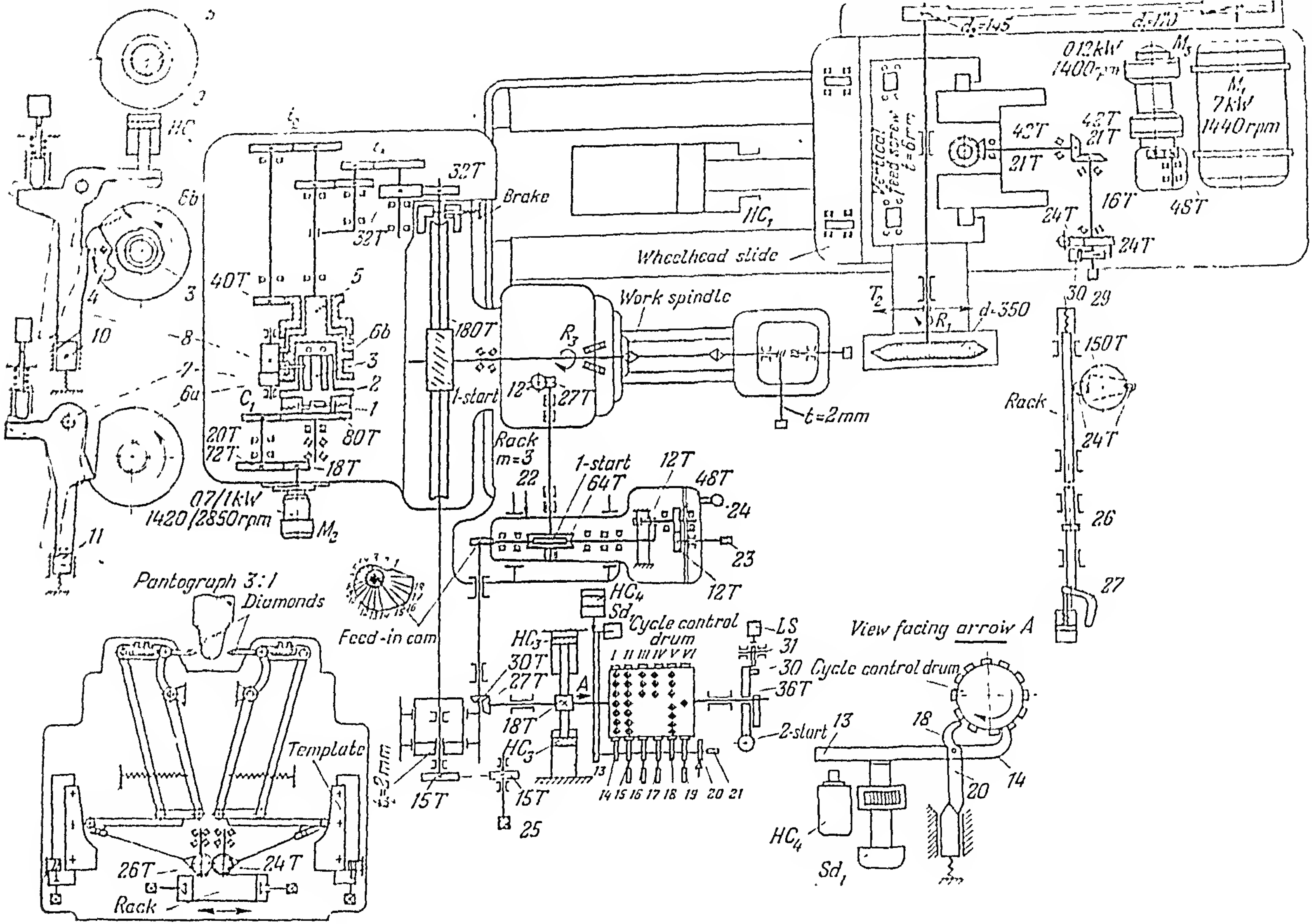

Fig. 57. Kinematic diagram of the semiautomatic gear-grinding machine model 586, made by the Moscow Machine Tool Plant, MSZ (structure class E33)

pawl I does not engage with ratchet wheel J since locking member 8 holds it in the retracted position. Locking member 8 not only locks disk $6b$, but it simultaneously locks time disk 9 mounted freely on indexing sleeve 5 and driven through change gears i_0. Hydraulic cylinder HC_4 can turn locking member 8 to disengage it from disks $6b$ and J.

When the wheelhead slide and grinding wheel leave the cutting zone special stops on the slide actuate hydraulic cylinder HC_2 which turns locking member 8. The latter disengages disks $6b$ and J and permits pawl I to engage with ratchet wheel J. At this index disks $6a$ and $6b$, time disk J and the work gear begin to rotate. When time disk 9 has made one full revolution, index disk $6b$ makes one, two, four or six revolutions depending upon the setup of change gears i_0. Since at this time the slot of disk $6b$ will be aligned with that of time disk 9, the locking member locks the two disks, retracting pawl I. At this the index disks stop rotating, the indexing cycle is concluded, the work gear will have been turned through $\frac{1}{z}$ revolution, and it will be possible to grind the next tooth space.

Locking members 7 and 8 are, in turn, registered by spring loaded slide blocks 10 and 11 to prevent them from taking some accidental position.

The indexing time can be selected in a range from 0.2 to 4 seconds depending upon the number of teeth to be ground; the maximum time is employed for small numbers of teeth. Such speed setting up for the indexing motion by means of separate time change gears i_0 is seldom applied, and then only in precision grinders, for example in heavy gear grinders. This device promotes more accurate and dependable operation of the indexing mechanism in grinding work gears with various numbers of teeth. If the hydraulic cylinder is not switched off the work spindle can be rotated continuously, such rotation being required in setting up the grinder to check the radial runout of the work gear. For this purpose electric motor M_4 is switched over to higher speed (2 800 rpm).

The work spindle can be rotated by hand to divide the allowance more uniformly on the profiles of two adjacent teeth. This is done by turning head 2. This rotation is transmitted through a chain drive $\frac{1}{15}$ to a screw with a pitch $t_4 = 2$ mm. The screw traverses a slide block which, in turn, shifts the worm of the worm gearing $\frac{1}{180}$ in the work head. Axial motion of the worm turns the worm wheel. A continuously applied brake retards the index worm wheel $180T$ to eliminate backlash and to ensure smooth indexing without jerks.

The kinematic group for the vertical infeed motion FI (T_4) is simple but it has several drives. The motion T_4 is not shown in the diagram since it is perpendicular to the plane of the diagram. Motion T_4 is accomplished

by the work head together with rack *12* which is rigidly mounted on the head.

If this motion is to occur automatically, it is powered by hydraulic cylinder HC_3, linked by a rack-and-pinion drive (pinion 18*T*) with the shaft of the cycle control drum. The piston of hydraulic cylinder HC_3 is continuously under pressure and tends to turn the cycle control drum, but is retarded in this by an escapement mechanism. The latter is made up of levers *13*, *14*, *18* and *20*, and the stops in rows *I* and *V* of the drum. All levers from *13* through *20* are rigidly mounted on shaft *21*. If hydraulic cylinder HC_4 or solenoid Sd_1 turns lever *13*, lever *14* will be withdrawn from the stops of the first row, and the cycle control drum will turn by the action of hydraulic cylinder HC_3 until the next stop of row *V* runs up against lever *18*.

The feed-in cam rotates together with the cycle drum. Depending upon the cam lobe being used (there are 18 in all), the feed-in cam shifts housing *22* of the feed-in mechanism and, with it, the worm of worm gearing $\frac{1}{64}$, a certain amount. Axial shift of the worm leads to rotation of worm wheel 64*T* and rack pinion 27*T*. This traverses rack *12* and, with it, the work head by a preset amount to remove the grinding allowance in roughing (sections *1* through *11* of the feed-in cam), finishing (sections *12* through *15*) or sparking-out operation (sections *16*, *17* and *18*). If grinding is being carried out with manual controls, rotation of head *23* is transmitted through a planetary drive with gears $\frac{12}{48} \diagup \frac{12}{72}$ and worm gearing $\frac{1}{64}$ to rack pinion 27*T*. To set the work head rapidly in the required position, handle *24* is turned, after disengaging gears 12*T* and 48*T*. Then worm gearing $\frac{1}{64}$ will rotate together with handle *24*.

Since the profile of the teeth being ground is obtained by the forming method, reproducing the profile of the formed grinding wheel, the grinder is furnished with three devices for truing and dressing the wheel: pantographs for side dressing with scale reductions of 3 : 1 and 6 : 1 (only the 3 : 1 pantograph is shown in the diagram), and a mechanism for dressing (and truing) the periphery of the wheel (not shown in the diagram). The 3 : 1 pantograph has a hydraulic drive; it is of commonly employed design and is for grinding gears of large module. Since the wheel is dressed in several passes, the wheelhead is vertically traversed by means of a screw with a pitch t_1 = 6 mm. This is an automatic motion. Upon longitudinal traverse of the wheelhead slide, when it is within the dressing zone, pin *26* runs up against sliding stop *27*. This pushes the pin upward together with rack *28*. The latter turns rack pinion 24*T* and segment gear 24*T* which carries a pawl. The latter turns ratchet wheel 150*T*; rotation is transmitted further through bevel gears $\frac{21}{42}$ $\frac{21}{42}$ to a vertical feed screw with a pitch t_2 = 6 mm. This motion

occurs automatically. Adjustable shield *30*, arranged above ratchet wheel *150T*, varies the number of teeth engaged by the pawl in one swinging motion, and thus changes the vertical infeed of the grinding wheel. The wheel can be set vertically by hand by turning shaft *29*.

This semiautomatic grinder has combination electro-hydromechanical controls. It can operate with manual controls for setting-up purposes and with a closed automatic cycle. In the latter case, the grinder is controlled by the cycle drum. This drum has four rows of variable stops (which can be screwed into the required holes of the drum). Up to 18 stops can be used. The drum also has two rows (*V* and *VI*) of fixed stops. Freely mounted on its shaft at the right side of the drum is worm wheel *36T*. Stop *20* on the end face of the worm wheel operates limit switch *LS* through lever *31* to transmit a command which returns the cycle drum to its initial position. By rotating the 2-start worm, the stop can be set to the beginning of the cycle from any type of grinding: roughing, finishing or sparking-out.

After setting the corresponding electrical devices to the AUTO CYCLE position and pressing the CYCLE START pushbutton, rough grinding of the teeth begins. In several passes and with several vertical infeed motions of the work gear, one tooth space is ground until the preset rough grinding allowance is removed. Then the work gear is indexed to the next tooth space and the cycle drum returns to its initial position. Shortened longitudinal travel of the wheelhead at the maximum feasible speed is automatically set up for rough grinding. At the end of the roughing operation, the grinding wheel is withdrawn by stops of row *IV* for dressing, while the stops of row *V* prepare the grinder for changing over to finish grinding, in which case the length of longitudinal travel increases and the traverse speed is reduced. The order in which the teeth are ground is also changed. The whole allowance for finish grinding is not removed all at one time from each tooth as in roughing; it is removed consecutively from all the teeth. Hence, indexing takes place after each full stroke (back and forth) of the wheelhead slide, and vertical infeed of the wheelhead takes place only after one full revolution of the work gear.

A partially automated cycle is also available. In this case, the work gear is rough ground in an automatic cycle and finished with manual controls.

Two change gear units i_x and i_0 are to be set up. Let us derive the corresponding setup formulas.

Indexing change gears i_x.

The basic displacements are

$$k \text{ revolutions of index disk } 6b \rightarrow \frac{1}{z} \text{ revolution of the work gear}$$

The kinematic balance equation is

$$k \cdot i_x \cdot \frac{12}{12} \cdot \frac{1}{150} = \frac{1}{z}$$

Hence the setup formula is

$$i_x = \frac{180}{kz} \tag{46}$$

Time change gears i_0

The basic displacements are

1 revolution of time disk *9* $\rightarrow k$ revolutions of index disk *6b*

Then the kinematic balance equation is

$$1 \times \frac{80}{40} \times \frac{1}{i_0} = k$$

and the setup formula is

$$i_0 = \frac{2}{k} \tag{46'}$$

The available numbers of revolutions of the index disk during one indexing cycle are: $k_1 = 1$, $k_2 = 2$, $k_3 = 4$ and $k_4 = 6$.

On the basis of the kinematic balance equation between motor M_2 and index disk *6b*, it is possible to determine the cycle time t_o for the various values of k. Thus

$$\frac{1{,}420}{60} \times \frac{18}{72} \times \frac{20}{80} \times i_0 \times \frac{40}{80} = n_o$$

but $t_o = \frac{1}{n_o}$ so that, finally, $t_o = \frac{48k}{71}$

$$\begin{array}{ll} \text{for } k_1 = 1 & t_1 = 0.7 \text{ sec} \\ \text{for } k_2 = 2 & t_2 = 1.4 \text{ sec} \\ \text{for } k_3 = 4 & t_3 = 2.7 \text{ sec} \\ \text{for } k_4 = 6 & t_4 = 4 \text{ sec} \end{array}$$

The minimum values of t_o are to be employed in grinding gears with large numbers of teeth.

5-6. Kinematic Structure of Bevel-Gear-Cutting Machines

Methods of Cutting Bevel Gears

Bevel gears are widely employed in mechanical engineering for transmitting rotation between intersecting shafts, as are hypoid gears for crossed shafts.

From the fundamental laws of toothed gearing it is known that the tooth profiles of bevel gears are spherical curves and thus are distinguished from involutes which are plane curves. Replacing the spherical surface on which

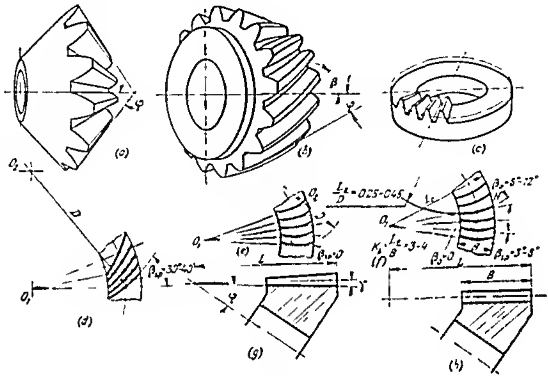

Fig. 58. Types of bevel gear teeth

the tooth profile should be constructed with a conical surface tangent to the sphere (back cone), we obtain octoid gearing which closely approximates involute gearing in shape.

As to the shape of the teeth along their length, bevel gears may be straight-tooth (Fig. 58*a*) or curved-tooth, with the teeth inclined at an angle β to the elements of the pitch cone (Fig. 58*b*).

The fact that the curves, determining the shape of the teeth along their length, are located on the surface of a cone makes it difficult, in the general case, to discover the laws governing the generation of such curves of double curvature. These laws are a vital necessity, however, in order to devise methods of profiling the teeth and to work out the kinematics of the corresponding machine tools. Because of this, the shape of the tooth of a bevel gear is usually determined by the form of the tooth of the conjugate (mating) crown gear. The latter is a bevel gear with a pitch angle of $\varphi = 90°$ (Fig. 58*c*), i.e. its pitch surface is a plane. It is easy to see that such a crown gear is actually a circular basic rack. It is the limiting bevel gear ($2\varphi = 180°$)

in the same sense that a rectilinear gear rack is the limiting spur or helical gear with a radius $R \to \infty$.

The lines on the circular, or crown, rack, determining the shape of the teeth along their length, lie in a plane and may be straight, circular arcs, prolate epicycloids or hypocycloids, prolate or curtate involutes, etc.

The use of these lines may be explained by the fact that teeth of the corresponding types can be machined by a combination of simple uniform motions—rotary and rectilinear—which are most readily produced in machine tools.

Of the curved teeth, most frequently employed is the circular-arc or, as it is usually called, spiral tooth with a spiral angle β ranging from 30° to 40° (Fig. 58*d*). Such teeth are easier to cut than curved teeth of other shapes and they can also be ground. Spiral bevel gears, as well as other curved-tooth bevel gears, cannot be applied in many cases because of the high axial loads, due to the fact that for such teeth $\beta \neq 0$.

In an attempt to reduce these axial loads, various versions of circular-arc teeth were proposed. One of these, the proprietary Zerol bevel gear has curved teeth whose spiral angle is equal to zero at the middle of the tooth (Fig. 58*e*). Zerol bevel gears, however, have not found wide application because their curved teeth have two branches—right and left. This requires precise alignment of the mating gears to obtain proper engagement; under load such gears operate, for the most part, with a single branch of the teeth. An improved tooth shape is one in which the circular-arc teeth have a spiral angle $\beta_o \simeq 0$ at the small diameter (Fig. 58*f*). The axial thrust of these gears is only 10 to 15 per cent of that in bevel gears having a spiral angle $\beta_{sp} = 35°$.

As a rule, the whole depth of both spiral and straight-tooth bevel gears decreases along the tooth length (Fig. 58*g*). Bevel gears with teeth of constant depth (Fig. 58*h*) are also used, however.

Internal bevel gears have found no application whatsoever, since they possess no advantages and are more difficult to manufacture than external bevel gears.

Sometimes, bevel gears are made with profile modifications such as semi-topping or toprem (top removal). Straight-tooth bevel gears may be crowned to provide localized tooth bearing.

The size range of bevel gears is narrower than that of spur or helical gears: diameters range from 5 to 2,000 mm and modules from 0.3 to 20 mm.

The shape of the tooth profile is not constant along the length of the tooth. The side surface of an involute tooth located on a cone is a more complex geometrical surface than that of spur or helical gear teeth. Hence, not all methods used in cutting spur and helical gears can be applied in the manufacture of bevel gears.

One method of low production capacity is used, though only rarely, in cutting very large straight-tooth bevel gears. It is based on the double tracing

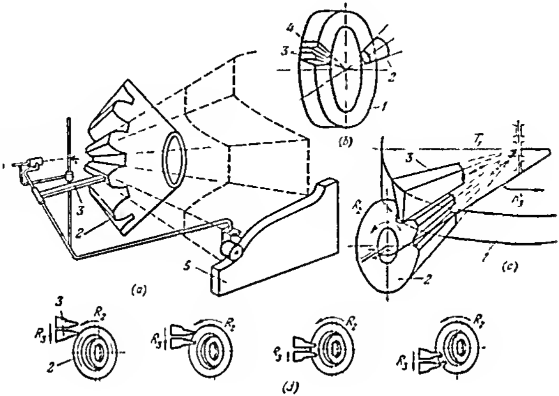

Fig. 59. Methods of cutting bevel gears

method (almost never used in cutting spur and helical gears) in which single-point tools 3 plane the teeth to a template 5 (Fig. 59a). The corresponding machines are called template-type bevel gear planers.

The forming and tangent method, in which the tooth surface is milled with a gear tooth milling cutter (form cutter), finds application only for rough cutting bevel gears. The use of this method for finishing the teeth involves considerable difficulties due to the variable shape of the profile along the length of the teeth. It has not found application except for makeshift repair jobs.

In contrast to the two preceding methods, the tracing and generating method, simply called the generating method, is extensively employed in the production of bevel gears of all types. Here, as in cutting spur and helical gears, the basic rack tooth contour is used as the generating contour, in the given case (1 in Fig. 59b) a circular rack, or crown gear, and its tooth,

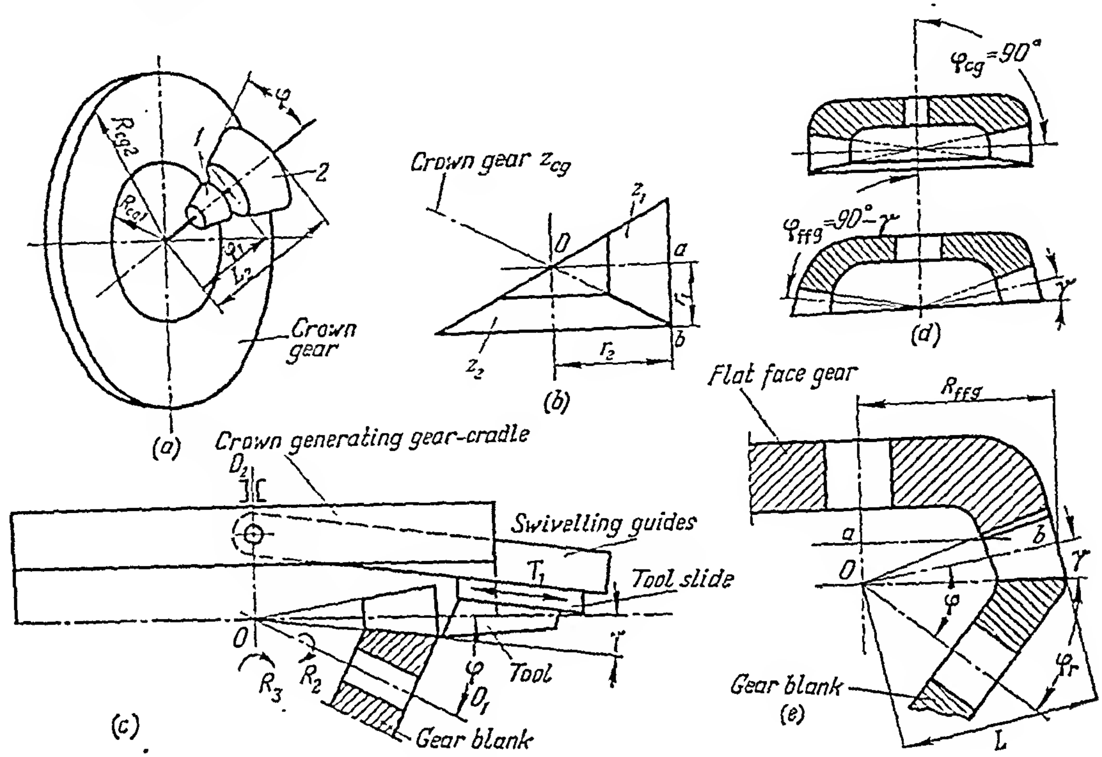

Fig. 60. Generating crown gear

or more precisely, its tooth space is represented by the straight cutting edges *4* of two special tools *3*. If a reciprocating motion T_1 is imparted to the generating contours *4* along the elements of a cone (Fig. 59*c*) at a speed equal to the required cutting speed, and gear blank *2* is slowly rolled around crown gear *1* by means of two interrelated rotary motions R_2R_3, the tools will cut a straight tooth on the blank. The successive stages in the generation of a tooth are shown in Fig. 59*d*. The rolling motion R_2R_3 of the gear blank in respect to the imaginary crown gear is accomplished, in this case as well, by two rotary motions: rotation R_3 of the crown gear with the cutting tools and rotation R_2 of the blank.

In producing the rolling operative motion, the ratio of the angular velocities of motions R_2 and R_3 remains constant and depends upon the ratio of the numbers of teeth z and z_{cg} of the gear being cut and the crown gear, respectively. In bevel gear generators, z_{cg} is a variable value depending upon the number of teeth z on the gear to be cut and its pitch angle φ.

It is evident from Fig. 60*a* that the radius R_{cg} of the crown gear is equal to the length L of an element of the pitch cone on the gear blank. The length L

is called the cone distance. Since the value L differs for various gears that are to be cut in the generator we can write

$$R_{cg1} = L_1 \quad R_{cg2} = L_2, \text{ etc.}$$

but

$$L = \frac{R_{gb}}{\sin\varphi} = \frac{mz_{gb}}{2\sin\varphi} \text{ and } R_{cg} = \frac{mz_{cg}}{2}$$

therefore

$$z_{cg} = \frac{z_{gb}}{\sin\varphi} \tag{47}$$

The number of teeth on the crown gear can be determined from another relationship as well. Since $R_{cg} = L = \sqrt{r_1^2 + r_2^2}$ (Fig. 60b) and

$$R_{cg} = \frac{mz_{cg}}{2} \quad r_1 = \frac{mz_1}{2} \text{ and } r_2 = \frac{mz_2}{2}$$

hence

$$z_{cg} = \sqrt{z_1^2 + z_2^2} \tag{48}$$

where z_1 and z_2 = numbers of teeth of the mating gears to be cut.

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OO_1 of the crown gear The angle φ is the pitch angle of the gear being cut. The direction of tool travel makes an angle γ (dedendum angle of the gear being cut) with the same plane For a single value of φ, angle γ may vary, depending on the module of the work gear. Therefore, the cradle of a generator designed on this principle must have swivelling guides for the tool slides enabling the generator to be set up to angle γ. This feature complicates the construction of the cradle and hence such cradles (with a crown generating gear) are rarely used in bevel gear generators.

To simplify the construction of the cradle (eliminating the swivel guides for the tool slides), instead of the ordinary crown gear (upper schematic drawing in Fig. 60d), a flat face gear (lower drawing) is used The latter has a face angle equal to 90° and a pitch angle φ_{ffg} $90° - \gamma$. In this case, the cradle is of simpler and more rigid design

The "meshing" of the gear to be cut and the flat face gear is shown in Fig. 60e. It follows from triangle Oab that

$$R_{ffg} - L\cos\gamma$$

and

$$z_{ffg} \quad \frac{z_{gb}\cos\gamma}{\sin\varphi} \tag{49}$$

L. Korostelev has shown that bevel gears cut in generators with a cradle based on a flat face gear are not actually conjugate. The distortion of the tooth profile, however, materially affects the operation of the bevel gearing only if the dedendum angle of the gear teeth exceeds 3°.

The kinematic structure of all machines for cutting bevel gears of either the straight- or curved-tooth type by the generating method must consist of two kinematic groups and, in some cases, of three, if a separate indexing group has been added. One group provides the formative roll motion R_1R_2 of the gear blank around the crown gear (Fig. 61a). This motion not only produces the tooth profile, but its shape along the tooth length as well, the latter depending upon the shape of the teeth on the crown gear which produces the second formative motion, called the motion for shaping the tooth along its length. We shall conditionally call the first motion the one for shaping the tooth profile.

In all bevel gear generators, the structure of the internal constraint in the group producing the first motion is the same (Fig. 61a), while the structure of the internal constraint of the other group may be simple, consisting of a single kinematic pair for rectilinear or rotary motion of the cutting tool, or complex. In the latter case, it is a multiple-link kinematic chain connecting two movable operative members, accomplishing elementary motions and producing a complex motion of the cutting tool in reference to the gear blank.

If a rectilinear reciprocating motion along the conical element of the gear blank is imparted to the cutting tool mounted on the generator cradle, a straight tooth will be obtained on the imaginary crown gear and on the bevel gear being cut. If this motion is directed at an angle β to the element of the pitch cone, a skew bevel gear is produced (Fig. 61b). When a face-mill type of cutter (with 18 to 24 blades) is used instead of two reciprocating tools, and a rotary motion R_3 about its axis is imparted to it, we obtain a curved tooth of circular-arc shape (Fig. 61c). By varying the position of the centre O_1 of the cutter in reference to the position of the centre O_2 of the crown gear (i.e. co-ordinates x and y), we can obtain various angles of inclination β of the teeth being cut. In both cases, the internal constraints are not to be set up kinematically.

Teeth with the shape of a prolate hypocycloid are produced if the cutter has a rolling motion. In this case, in addition to rotation R_3 about its own axis O_1, rotation R_4 about the axis O_2 of the crown gear is imparted to the cutter (Fig. 61d). When a circle of radius r rolls without slipping inside a circle of radius R, and the blades are at a distance of $O_1E > r$ from centre O_1, then each blade describes a prolate hypocycloid, a part of which is employed as the tooth side line. If a small circle rolls without slipping around the outside of a large circle of radius R, a tooth with a side line in the shape of a prolate epicycloid is produced. Both curves are employed in practice;

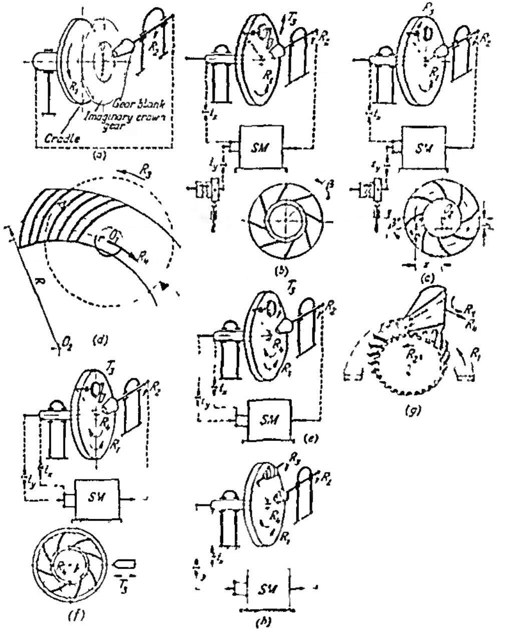

Fig. 61. Internal kinematic constraints in the formative groups of various bevel gear generators

the values of r and R are selected such that the like branches of the cycloid curve are spaced at distances of one or several whole circular pitches of the gear being cut.

The complex roll motion of the cutter in reference to the crown gear is produced by the internal kinematic chain connecting the cutter spindle with the crown gear spindle (Fig. 61*e*). This kinematic chain must have setting-up facilities. The basic displacements of the final members are: 1 revolution of the cutter $\rightarrow \frac{z_i}{z_{cg}}$ revolution of the crown gear, if the cutter has only one group of blades machining the same tooth; and $\frac{1}{q}$ revolution of the cutter $\rightarrow \frac{z_i}{z_{cg}}$ revolution of the crown gear, if the cutter has q groups of blades. Here z_i is the number of teeth skipped in consecutively cutting the teeth of the gear, and may be any whole number not having any common factors with the number of teeth on the gear to be cut.

To produce z_{cg} teeth on the crown gear, the cradle would have to make several revolutions, but the blades would not be in the cutting zone for sufficient time, to operate efficiently. Therefore, rotation of the crown gear is substituted by the corresponding rotation of the gear blank:

$$1 \text{ revolution of the cutter} \rightarrow \frac{z_i}{z_{cg}} \times \frac{z_{cg}}{z_{gb}} \text{ revolution of the gear blank}$$

or, finally,

$$1 \text{ revolution of the cutter} \rightarrow \frac{z_i}{z_{gb}} \text{ revolution of the gear blank}$$

A curved tooth can also be produced by two reciprocating tools. If the reciprocating motion T_3 of the tools, powered through a crank drive, is connected by a positive kinematic constraint with rotation R_4 of the cradle (Fig. 61*f*), the side line of the teeth on the crown gear will be a circular sinusoid.

The basic displacements are:

$$1 \text{ full stroke (back and forth) of the tool} \rightarrow \frac{z_i}{z_{cg}} \text{ revolution of the crown gear}$$

or

$$1 \text{ full stroke of the tool} \rightarrow \frac{z_i}{z_{gb}} \text{ revolution of the gear blank}$$

The notation here is the same as in the preceding case.

Shown schematically in Fig. 61*g* is the cutting of a curved-tooth bevel gear with a conical hob. In addition to rotation R_3 about its own axis, the hob also has rotation R_4 about the axis of the crown gear with the following

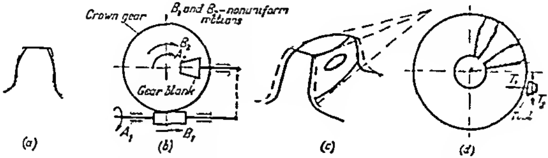

Fig 62 Diagrams of motions in cutting bevel gears with a modified profile or shape along the tooth length

ratio of speeds of these two rotary motions one revolution of the cutter in motion R_3 corresponds to $\frac{k}{z_{cg}}$ revolution of the cutter in motion R_4, where k is the number of starts on the hob With the aid of this second motion an involute shape will be produced along the length of the teeth on the crown gear Actually, the generating contour (tooth of the circular rack) participates in two motions a translatory motion along the hob axis (due to its rotation R_3) and a rotary motion R_4 about the axis of the crown gear. Thus, it reproduces the motion of a straight line rolling without slipping about a circle In such a motion, any point of the straight line describes an involute of a circle

The teeth obtained by this process differ from other curved teeth in that their thickness and height are constant over their whole length Figure 61h illustrates the internal constraints between the hob, gear blank and crown gear required to produce involute curves along the profile and length of the gear teeth

The internal constraints of the formative groups are altered to some extent if the initial shape of the profile or shape along the tooth length is modified To produce a tooth profile with a cut away tooth face (a modification known as topping) as in Fig 62a nonuniform rotation is imparted to the crown gear To this end the internal kinematic chain is made more complex by including an addi h shifts the index worm of the generate bevel gears, i e gears with localized re obtained by providing slightly reduced thickness at the ends of the teeth In this case the group for shaping the teeth along their length must be complex so as to produce a curvilinear motion This motion is composed of two interrelated rectilinear nonuniform motions T_1 and T_2 (Fig 62d)

The complete kinematic structure of these gear generators depends upon the method employed to interconnect the indexing and feed groups.

Several typical kinematic arrangements of bevel gear generators will be considered in the following.

Semiautomatic Spiral Bevel Gear Generator, Model 5A27C4

This generator is used for cutting spiral bevel gears of a diameter up to 500 mm (at a gearing ratio of 10 : 1) and module up to 10 mm. The machine can be expediently employed in all branches of the engineering industries, both for small- and large-lot production.

The kinematic structure of the generator (Fig. 63) consists of two kinematic groups for the formative motions F_v (R_1) and F_s (R_2R_3).

The group of the cutting motion F_v (R_1) is simple. Its internal constraint consists of a single rotary kinematic pair, made up of the cutter spindle and the cradle housing. The external constraint, consisting of the kinematic chain between motor M_1 and the cutter spindle, contains change gears i_v by means of which the cutting motion F_v (R_1) is set up in respect to velocity and direction.

The setup formula for change gears i_v is derived from the kinematic balance equation

$$1{,}440 \times \frac{12}{41} \times \frac{45}{45} \times i_v \times \frac{27}{27} \times \frac{23}{30} \times \frac{23}{30} \times \frac{58}{43} \times \frac{24}{24} \times \frac{17}{17} \times \frac{17}{91} = n$$

then

$$i_v = \frac{n}{62.5} \tag{50}$$

where n = speed of the face-mill type cutter, rpm.

The cutting speed gear train turned out to be quite long in this generator. This derives from the fact that the spindle can be set at various angles in respect to the axis of cradle rotation, and that it proves more expedient to transmit motion to the inclined spindle through rigid members and not through a universal joint that cannot transmit high loads.

To enable the spindle to be set to the specified angle of inclination in the required plane, the cradle is made up of several parts: cradle housing, eccentric, drum and spindle housing. The spindle housing contains a sleeve in which the spindle runs in antifriction bearings. The axis of spindle housing rotation is inclined to the axis of cradle rotation, as a result of which the spindle is set at an angle. The spindle housing is rotated manually by turning shaft A. At this, bevel gear $24T$ rolls around another bevel gear $24T$. The drum, rotated by turning shaft B, sets up the plane of inclination of the spindle. Upon drum rotation, bevel gear $43T$ rolls around bevel gear $58T$.

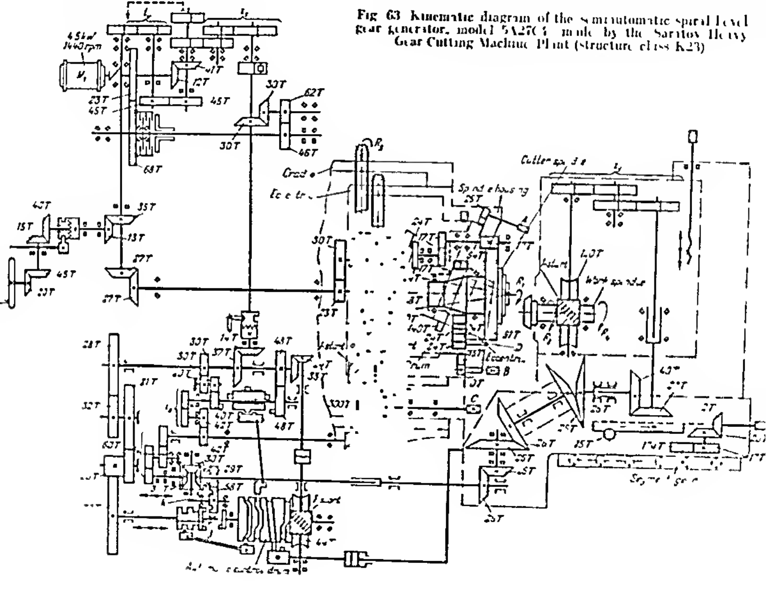

Fig 63 Kinematic diagram of the semiautomatic spiral bevel gear generator, model 5A27C1 made by the Saratov Heavy Gear Cutting Machine Plant (structure class K23)

Rotation of the eccentric, by turning shaft C, sets up the radius vector of the spindle axis position in respect to the cradle axis. At the same time, gear $30T$ rolls around gear $23T$. Turning shaft D by hand rotates the threaded spindle sleeve to set the cutter spindle axially.

Inclination of the axis of cutter rotation enables gears with various parameters to be generated with a cutter having blades of a single number (size). As a result, a considerably smaller set of cutters is required to cut the whole range of gears within the capacity of the machine. Moreover, spindle inclination enables the gear of Formate gearing to be cut with a normal bevel generating gear instead of a flat face gear, thereby eliminating the need for a complex and expensive modified roll attachment. Formate gearing is the name given to mating pairs in which the gear member (larger of the pair) has straight-sided teeth and is produced by the forming method (without generation). The pinion (smaller of the pair) is mated to the gear by the generating method so that it meshes properly with the straight profiles of the gear member.

The second kinematic group, for the feed motion F_s (R_2R_3) is complex; the internal kinematic constraint between the cradle and work spindle consists of a single kinematic profiling gear train with two change gear units i_x and i_y.

The external constraint of the group consists of the drive train which transmits motion from motor M_1 through change gear units i_s and i_0 to the internal constraint. Thus, the feed motion group includes four setting-up devices, two of which are in the drive train. Change gears i_y are located in the common branch which belongs to both the profiling and indexing gear trains, and is used to set up the kinematic group for indexing. Thus, the operative motion F_s (R_2R_3) is to be set up in respect to three kinematic parameters: change gears i_x set up the path, change gears i_0 set up path length, and change gears i_s set up the velocity of the motion.

Three kinematic chains must be considered to derive the setup formulas for these change gears.

Profiling gear train (*with change gears* i_x). As a rule, the following basic displacements are transmitted to the final members of this train:

$$1 \text{ revolution of the cradle} \longrightarrow \frac{z_{cg}}{z_{gb}} \text{ revolutions of the work spindle}$$

The kinematic balance equation is

$$1 \times \frac{300}{1} \times \frac{16}{32} \times i_x \times 1 \times$$
$$\times \frac{26}{26} \times \frac{26}{26} \times \frac{26}{26} \times \frac{40}{20} \times i_y \times \frac{1}{120} = \frac{z_{cg}}{z_{gb}}$$

Hence, after substituting the value of the change gear ratio i_y [illegible] (see below), we obtain for the profiling change gears

$$i_x = \frac{z_{c_0}}{z_0} \qquad (51)$$

Cradle roll gear train (with change gears i_0) Let us denote the angle of cradle roll during working travel (the process of actually cutting a tooth) by θ One final member of this train is the cradle, the other is the automatic control drum which operates the reversing device and thereby terminates the duration of working travel A fixed cam on the drum actuates the cradle reversing member Consequently, the duration of forward and reverse cradle roll depends upon the ratio of cradle rotation speeds in one direction and in the other It is evident from the gearing ratios of the reversing mechanism gears that the speed of forward rotation is one half of that of reverse rotation Hence, the basic displacements are the following displacements of the final members of this gear train

$$\frac{2}{3} \text{ revolution of the automatic control drum} \rightarrow \frac{\theta}{360} \text{ revolution of the cradle}$$

However, these basic displacements should be refined During reversal the cradle stops while the automatic control drum continues to rotate Therefore, we shall determine the number of revolutions n_{or} of the output shaft of the reversing device during working travel of the cradle, i e during $\frac{2}{3}$ revolution of the automatic control drum

$$n_{or} = \frac{2}{3} \times \frac{[illegible]}{1} \times \frac{2[illegible]}{31} \times \frac{30}{[illegible]0} \times \frac{20}{[illegible]0} = 10\ 66 \text{ revolutions}$$

It is evident that the reversing device is switched over after 10 revolutions of its driven shaft Consequently the true basic displacements kinematic balance equation and the setup formula are

$$10 \text{ revolutions of the driven shaft in the reversing device} \rightarrow$$
$$\rightarrow \frac{\theta}{360} \text{ revolution of the cradle}$$

Hence

$$10 \quad i_0 \quad [illegible] \quad \frac{[illegible]}{1} \times \frac{1}{300} = \frac{\theta}{300}$$

and finally

$$i_0 = \frac{\theta}{[illegible]} \qquad ([illegible]2)$$

Feed gear train (with change gears i_s). The cycle feed, or cycle time, commonly employed in bevel gear generators, is defined as the time in seconds required to machine one tooth. One tooth is cut during one revolution of the automatic control drum, and the time t_c sec, selected to suit the cutting conditions, can be expressed as the number of motor shaft revolutions during this time. Therefore, the basic displacements of the final members can be written as

$$1 \text{ revolution of the automatic control drum} \rightarrow$$

$$\rightarrow \frac{1{,}440}{60} t_c \text{ revolutions of the motor shaft}$$

In distinction to the previous model (5A27C1), this generator has a rapid traverse mechanism which can transmit rapid motion to the internal constraint through an overrunning clutch, bypassing the feed change gears i_s. As a consequence, the drum rotates at two different speeds while the time of the reverse roll of the cradle remains constant and equal to 3.85 sec. Therefore the time required for the whole cycle $t_c = t_w + t_i$ cannot be used in the basic displacement equations. Only the working travel time t_w in seconds can be employed. Then the basic displacements are

$$\frac{2}{3} \text{ revolution of automatic control drum} \rightarrow$$

$$\rightarrow \frac{1{,}440}{60} t_w \text{ revolutions of the motor shaft}$$

The kinematic balance equation and setup formula are:

$$\frac{2}{3} \times \frac{44}{1} \times \frac{24}{33} \times \frac{37}{14} \times \frac{1}{i_s} \times \frac{41}{12} = \frac{1{,}440}{60} t_w$$

and

$$i_s \cong \frac{8}{t_w} \tag{53}$$

In addition to the two formative groups, the machine also has a group for the indexing motion *Ind* (R_4). The indexing process is carried out on the principle of parallel indexing through a differential.

The indexing motion R_4 is periodic. It is engaged and disengaged in the following way.

Upon the reverse roll of the cradle, the automatic control drum, through a lever system, shifts the rotating driving member j of a Geneva wheel mechanism until its rollers engage and begin to rotate the driven member k of this mechanism. The two rollers are arranged on disk j at the same radius and at 90° from each other. Member k has four slots. Thus in one revolution

of disk j, Geneva wheel k makes half a revolution and turns the differential housing one revolution. After this the indexing motion ceases since the rollers of disk j run out of the slots of wheel l.

Hence, the basic displacements are

$$1 \text{ revolution of driving disk } j \text{ of the Geneva wheel} \rightarrow$$

$$\rightarrow \frac{1}{z_b} \text{ revolution of the work spindle}$$

The corresponding kinematic balance equation is

$$1 \times \frac{1}{2} \times \frac{48}{24} \times 2 \times \frac{24}{z_b} \times \frac{24}{24} \times \frac{24}{24} \times \frac{1}{20} \times i_y \times \frac{1}{120} = \frac{1}{z}$$

and the setup formula is

$$i_y = \frac{40}{z_b} \qquad (51)$$

The gear blank is withdrawn from the cutter in two ways. Automatic withdrawal is effected by cam curves of the automatic control drum just preceding the indexing process. The sliding base with the indexing head can be retracted at any time by means of a hydraulic cylinder arranged between the drum and carriage. Oil is admitted to this cylinder through a manually controlled valve.

The automatic control drum has two cam curves for withdrawing the gear blank. One curve, as mentioned above, serves for withdrawing the gear blank from the cutter before the beginning of the indexing process and then re-advances the blank to the initial position before the beginning of the profiling process. The other curve (the extreme right hand one on the drum) is used in roughing gears when tooth spaces with straight sides, i.e., of trapezoidal shape (reproducing the shape of the cutter blades), are cut without using the generating motion.

The second curve, more gradual than the first, effects slow traverse of the sliding base with the indexing head, advancing the blank to the cutter and thereby accomplishing the feed-in motion. When the cutter blades reach the bottom lands of the tooth spaces, the blank is rapidly retracted and indexed. During the roughing operation the cradle [illegible] does not rotate. The blank rotates only [illegible] space. The forming method [illegible] than generating since the path length [illegible] mately to the tooth depth [illegible] generation especially in cutting [illegible]

Hypoid gears can also be cut [illegible] spindle is set at the required height

Straight Bevel Gear Generator, Model BF-201A (Klingelnberg Co., FRG)

This generator (Fig. 64) is intended for cutting straight-tooth bevel gears with a cone distance up to 139 mm, of a diameter up to 260 mm for a gearing ratio of the mating pair up to 1 : 5 and with a module up to 10 mm. The gears are cut by generation with two rotating circular cutters.

This machine differs from similar models in that it is designed for large-lot or mass production and its construction is highly suited for this purpose. Its rotating cradle carries the gear blank instead of the cutting tools. This arrangement enables a more rigid and powerful drive to be provided for the cutting tools, thereby increasing the output of the generator.

Instead of using the usual reciprocating tools, the straight bevel-gear teeth are cut by the generating principle with disk type cutters of large diameter (up to 600 mm). The cutters have no longitudinal travel along the tooth length. The indexing group in this machine is interconnected with the feed group in series. Thus, an interchangeable indexing clutch with a driving gear *a* having 36, 48 or 72 teeth is installed in the indexing gear train. This clutch operates in the following way. After cutting one tooth, while the cradle returns to its initial position, the indexing clutch is disengaged and gear *a* does not rotate until gear 60*T* of change gears i_x makes one complete revolution. During this time the gear blank is stationary, only the cradle rotates. Then the clutch engages and the generating (roll) motion is resumed.

The kinematics of this generator are simple in essence, but due to the unique arrangement, there is a hidden planetary drive in the feed gear train, since gear 38*T* rolls around gear 110*T* during cradle rotation. The worm wheel of the cradle is stationary, while the worm simultaneously rotates both about its own axis and about the axis of the cradle. Thus the worm, having two degrees of freedom, converts the worm gearing into a summation mechanism. Since the hidden planetary drive is within the external constraint, it does not distort the shape of the teeth and only changes the rate of feed.

Next we shall derive the setup formulas.

1. *Indexing change gears* i_x

$$1 \text{ revolution of gear } 60T \longrightarrow \frac{1}{z_{gb}} \text{ revolution of the gear blank}$$

$$1 \times i_x \times i_0 = \frac{1}{z_{gb}}$$

$$i_x = \frac{1}{i_0 z_{gb}} \tag{55}$$

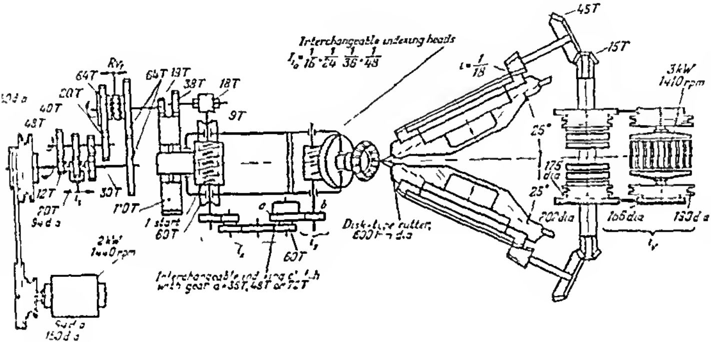

Fig. 61 Kinematic diagram of the straight bevel gear generator, model BF-201A, made by the Klingelnberg Co., FRG (structure class C23)

2. *Profiling change gears* i_y

$$1 \text{ revolution of the cradle} \rightarrow$$

$$\rightarrow \frac{z_{cg}}{z_{gb}} \text{ revolutions of the gear blank}$$

$$1 \times \frac{60}{1} \times i_x \times i_y \times i_0 = \frac{z_{cg}}{z_{gb}}$$

$$i_y = \frac{z_{cg}}{60} \tag{56}$$

3. *Cutting speed train*

$$1{,}410 \times i_v \times \frac{15}{45} \times \frac{1}{18} = n_c$$

and

$$i_v = \frac{9}{235} n_c \tag{57}$$

where n_c = cutter speed, rpm.

4. *Feed train.* Cycle feed is employed in this as in other similar machines; it is determined by the time t_w in seconds required for the working roll (travel) of the cradle. The working roll, in turn, is determined by the amount, or angle, of cradle roll. The basic displacements are

$$\frac{1{,}440}{60} t_w \text{ revolutions of the motor shaft} \rightarrow$$

$$\rightarrow \frac{\theta_{gb}}{360} \text{ revolution of the cradle}$$

then

$$\frac{1{,}440}{60} \times t_w \times i_1 \times i_s \times \frac{20}{64} \times \left(\frac{19}{110} \times \frac{110}{38} \pm \frac{110}{38}\right) \times \frac{9}{18} \times \frac{1}{60} = \frac{\theta_{gb}}{360}$$

Assigning a value of i_1, the ratio i_s is determined. Then the gears in the feed gearbox are shifted to comply with this ratio.

Semiautomatic Straight Bevel Gear Generator, Model 5250

This generator (Fig. 65) is intended for cutting straight-tooth bevel gears up to 500 mm in diameter and with a module up to 8 mm, of both the conventional and crowned types with reciprocating planing tools.

A special attachment is available for planing spiral bevel gears; thus the machine is suitable for repair operations. The generator differs from earlier models in that the indexing group is connected to the feed group by means of the compound group interconnection principle.

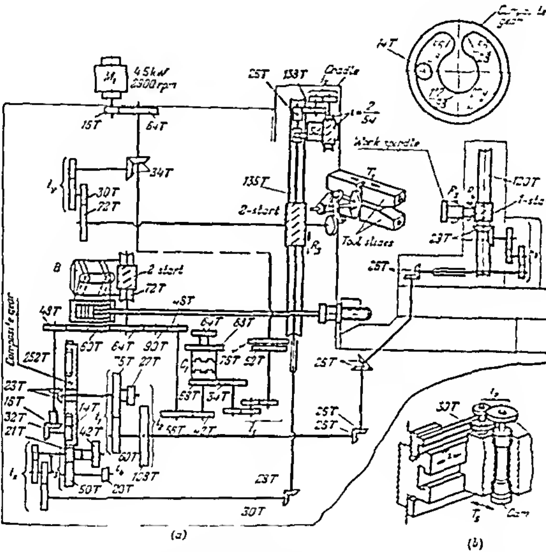

Fig. 65. Kinematic diagram of the semiautomatic straight-bevel gear generator, model 5250, made by the ENIMS Stankokonstruktsiya Plant of Moscow (structure class K2?)

Let us consider the structure of the generator as set up for producing crowned straight tooth bevel gears. In this case, the machine has the following operative motions: cutting motion F_v (T_1T_2), feed motion F_s (R_2R_3) and the indexing motion $Ind\,(R_4)$

The group of the cutting motion F_v $(T_1\ T_5)$ is complex and consists of the internal kinematic constraint which connects longitudinal motion T_1 through rack pinion $30T$, change gears i_0 and a cam with the axial motion T_5. The drive transmits motion from motor M_1 to the tool slides.

The feed motion F_s (R_2R_3) is produced by the internal constraint connecting cradle rotation R_3 with gear blank rotation R_2 through profiling change gears i_x, indexing change gears i_y and the composite gear of the special reversing mechanism. The external constraint of the feed group transmits motion from motor M_1 through feed change gears i_s and a series of gears to the bevel gearing $\frac{23}{23}$. From these bevel gears motion is transmitted to the right through change gears i_y to the work spindle, and from the same gears to the left through the reversing mechanism and change gears i_x to the cradle.

The special reversing mechanism, based on the composite gear, only reverses the cradle; the gear blank continues to rotate in the same direction. In this mechanism, driving gear $14T$ rotates continuously in a single direction, meshing with the internal section $224T$ of the composite gear. If the driving gear rotates counterclockwise, the composite gear also rotates counterclockwise. In this rotation, the left-hand internal half-gear $56T$ approaches driving gear $14T$ which it engages. The driving gear, however, is mounted in a slide that can move toward the axis of rotation of the composite gear. If this slide is released, gear $14T$ rolls around the half-gear and engages the external section $112T$ of the composite gear which then begins to rotate in the clockwise direction. Then the right-hand half-gear approaches the driving gear $14T$ from underneath, the latter rolls around the half-gear and then the composite gear reverses again.

The group for the indexing motion Ind (R_4) is simple. Its internal constraint is the rotary kinematic pair made up of the work spindle and the indexing (work) head. The drive to the indexing group is the same as that in the feed group, but to the latter a part of the internal constraint of the feed group is added.

Rapid rotation of the cradle and gear blank is effected by two gear trains containing the sliding cluster gear $52T$ and $76T$ located in the feed train near clutch C_1.

Bevel gears are roughed in this generator by the forming method, cutting teeth with straight sides, instead of the generating method. In this case it is necessary to apply radial infeed which is effected by the aid of a special curve on the automatic control drum B. This curve has a gradual part for slowly feeding the tool to the full depth of the tooth space, and a steeper part for rapid retraction of the tool. In addition, the gear blank is not indexed through several teeth z_i, as in generation, but consecutively tooth after tooth. To this end, two sliding gears are provided in the profiling gear

The group of the cutting motion F_v $(T_1\ T_5)$ is complex and consists of the internal kinematic constraint which connects longitudinal motion T_1 through rack pinion $30T$, change gears i_0 and a cam with the axial motion T_5. The drive transmits motion from motor M_1 to the tool slides.

The feed motion F_s (R_2R_3) is produced by the internal constraint connecting cradle rotation R_3 with gear blank rotation R_2 through profiling change gears i_x, indexing change gears i_y and the composite gear of the special reversing mechanism. The external constraint of the feed group transmits motion from motor M_1 through feed change gears i_s and a series of gears to the bevel gearing $\frac{23}{23}$. From these bevel gears motion is transmitted to the right through change gears i_y to the work spindle, and from the same gears to the left through the reversing mechanism and change gears i_x to the cradle.

The special reversing mechanism, based on the composite gear, only reverses the cradle; the gear blank continues to rotate in the same direction. In this mechanism, driving gear $14T$ rotates continuously in a single direction, meshing with the internal section $224T$ of the composite gear. If the driving gear rotates counterclockwise, the composite gear also rotates counterclockwise. In this rotation, the left-hand internal half-gear $56T$ approaches driving gear $14T$ which it engages. The driving gear, however, is mounted in a slide that can move toward the axis of rotation of the composite gear. If this slide is released, gear $14T$ rolls around the half-gear and engages the external section $112T$ of the composite gear which then begins to rotate in the clockwise direction. Then the right-hand half-gear approaches the driving gear $14T$ from underneath, the latter rolls around the half-gear and then the composite gear reverses again.

The group for the indexing motion Ind (R_i) is simple. Its internal constraint is the rotary kinematic pair made up of the work spindle and the indexing (work) head. The drive to the indexing group is the same as that in the feed group, but to the latter a part of the internal constraint of the feed group is added.

Rapid rotation of the cradle and gear blank is effected by two gear trains containing the sliding cluster gear $52T$ and $76T$ located in the feed train near clutch C_1.

Bevel gears are roughed in this generator by the forming method, cutting teeth with straight sides, instead of the generating method. In this case it is necessary to apply radial infeed which is effected by the aid of a special curve on the automatic control drum B. This curve has a gradual part for slowly feeding the tool to the full depth of the tooth space, and a steeper part for rapid retraction of the tool. In addition, the gear blank is not indexed through several teeth z_i, as in generation, but consecutively tooth after tooth. To this end, two sliding gears are provided in the profiling gear

train One gear is adjacent to change gears i_x and has 42 teeth while the other has 27 teeth This same method is used to cut the gear (larger) member of Formate pairs

A gear with straight sided teeth, as in Formate gearing, can be cut much faster by the forming method than by generation The pinion (smaller) member of the Formate pair is cut by generation, but with nonuniform rotation of the cradle This is accomplished by means of nonuniform axial shift of the worm in the cradle worm gearing $\frac{2}{135}$ Axial shift of this worm is effected by the action of eccentric cam E_1 The latter is driven from the worm of cradle worm gearing $\frac{2}{135}$ through gearing $\frac{26}{138}$, modified roll change gears i_z and worm gearing $\frac{2}{54}$ This mechanism is called the modified roll device and, when it is employed, the cradle worm gearing $\frac{2}{135}$ operates as a summation mechanism Two rotary motions are transmitted to the cradle worm wheel uniform motion due to rotation of the worm, and nonuniform motion due to axial shift of the worm Special calculations are required to set up the modified roll change gears i_z*

To set up the kinematics of the machine, it is necessary to determine the gearing ratios i_y, i_x, i_v and i_s

Indexing change gears i_y

$$1 \text{ revolution of drum } B \longrightarrow \frac{z_i}{z_{gb}} \text{ revolution of the gear blank}$$

then

$$1 \times \frac{72}{2} \times \frac{64}{48} \times \frac{23}{23} \times \frac{75}{60} \times \frac{26}{26} \times \frac{26}{26} \times \frac{26}{26} \times i_y \times \frac{29}{29} \times \frac{1}{120} = \frac{z_i}{z_{gb}}$$

and

$$i_y = 2\frac{z_i}{z_{gb}} \tag{58}$$

Profiling change gears i_x

$$1 \text{ revolution of the cradle} \times \frac{135}{2} \times \frac{28}{30} \times \frac{1}{i_x} \times \frac{21}{252} \times \frac{224}{14} \times \frac{32}{16} \times$$

$$\times \frac{23}{23} \times \frac{75}{60} \times \frac{26}{26} \times \frac{26}{26} \times \frac{26}{26} \times i_y \times \frac{29}{29} \times \frac{1}{120} = \frac{z_{cg}}{z_{gb}}$$

Hence

$$i_x = \frac{7}{2} \times \frac{z_i}{z_{cg}} \tag{59}$$

* The setup formula for the modified roll change gears was derived by V N Kedrinsky D Sc (Eng) and can be found in Kedrinsky and Pismannik *Bevel Gear Generators* Mashgiz Publishers Moscow, 1960 (Russ ed)

There are, however, machine tools with kinematic structures in which the velocity ratios of the elementary motions do not remain constant.

The structural arrangements of machine tools, accomplishing complex operative motions made up of two elementary interrelated motions having a variable velocity ratio, are shown in Fig. 66.

In such machine tools the complex formative motions are made up of several elementary motions, but one of these must necessarily be nonuniform. The velocity of this motion varies according to a mathematic expression dictated by the conditions of shaping the required surface. For example, in turning a circular cone, or taper (Fig. 66*a*), and a contoured surface of revolution (Fig. 66*b*) with a single-point tool, the same operative motions are employed, one of them, T_2T_3, being complex. In the first case (Fig. 66*a*), this motion is made up of two elementary uniform motions, or of two nonuniform motions, but such that the ratio of the instantaneous velocities of these motions is a constant value. In the second case (Fig. 66*b*), motion T_3 is nonuniform while motion T_2 is uniform and the ratio of the instantaneous velocities of these motions must necessarily be a variable value. From this it follows that the composition of the kinematic groups has not been altered, only the setting-up parameters have been changed. With the aid of a setting-up device in the internal constraints, it is necessary to obtain different laws of velocity variations of one of the elementary motions which make up the complex operative motion.

The setting-up devices used in these machine tools may be of various designs. In tracer-controlled lathes (Fig. 66*b*) and profile milling machines (Fig. 66*c*), in which the complex operative motion is made up of elementary rectilinear motions, interchangeable templates are employed in most cases. In these machines the working curve (profile) of the template provides for the relationship of the nonuniform cross displacement T_3 of the cutting tool and the uniform longitudinal displacement T_2. Thus

$$L_{T_3} = f(L_{T_2})$$

where L is the path length of the motion.

Velocity variation of a rotary motion can also be set up by means of change gears. The need for this may arise, for example, in using an ordinary gear shaper for cutting noncircular spur or helical gears with a special noncircular shaping cutter (Fig. 66*d*). Upon uniform rotation of the gear blank (motion R_1) it is necessary to transmit nonuniform rotation R_2 to the cutter. This is possible if noncircular gears are installed as change gears. Such noncircular gears are seldom employed because they are much more difficult to manufacture than circular gears. A gear hobber for cutting noncircular gears is considered below.

In most cases, nonuniform rotary motion is obtained by adding uniform and nonuniform motions, using a summation mechanism and an addition-

al internal kinematic train whose purpose is to convert uniform motion into nonuniform motion Such a structural arrangement (Fig 66e) has been employed in the spiral bevel gear generator, model 528, for cutting Formate gears with circular arc teeth with a face mill type of cutter It is evident from the diagram that the nonuniform rotary motion R_1 of the crown gear is due to two motions of the cradle worm uniform rotation and nonuniform axial travel (Fig 66e) The latter is transmitted to the worm by a crank disk The setting up procedure for varying the ratio of the velocities of motions R_1 and R_2, and for setting the limits of this ratio, consists of installing the necessary change gears and changing the crank radius The internal constraint between the work spindle and the crown gear consists of two internal kinematic chains The first of these consists of work spindle → → chango gears i_y → *summation* mechanism → change geare i_x → worm gearing and the crown gear In the second kinematic chain we have work *spindle* → *change gears* i_y → *summation mechanism* → *change gears* i_z → → crank mechanism → worm gearing and the crown gear In the latter case, the worm *gearing* operates *as a rack* and *pinion drive* The kinematic structure of a machine tool is much more complicated when nonuniform motion is obtained by summation mechanisms

We shall consider as examples a gear hobber for cutting *noncircular* gears and a machine for cutting threads with *nonuniform* pitch

Methods of Cutting the Teeth of Noncircular Gears

Noncircular gears employed in certain machinery and instruments may have various shapes, they may have a closed (complete) or open (incomplete) contour (Fig 67) The pitch curve of a noncircular gear may have the form of an ellipse, oval or *any* other line composed of both convex and concave parts of plane curves Transmissions with noncircular gears are used in textile and printing machinery, in computers and other mechanisms in which it is necessary to produce a nonuniform motion whose velocity conforms to a definite mathematical expression

The teeth of such gears have involute profiles, the pitch is constant, but the angle γ formed by the normal NN at a point on the pitch curve and

ariable value (Fig 67b) Variable

l the length of the radius vectors r

const and r = const) This com

a great extent, especially because

the variation in angle γ changes the initial position of the generating contour in respect to the radius vectors

If a noncircular gear is cut with a disk type gear milling cutter (Fig 67c), the indexing process will require, not only turning the gear blank through unequal angles from tooth to tooth, but shifting the blank as well to the

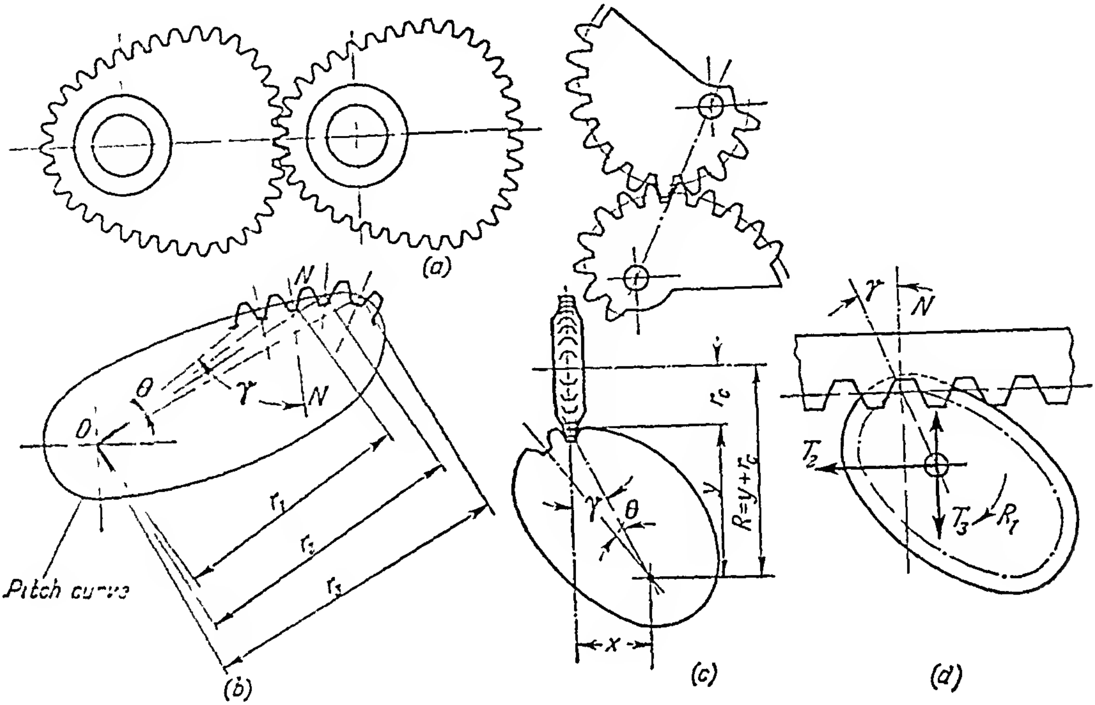

Fig. 67. Noncircular toothed gears

co-ordinate dimensions x and y (corresponding to the axis of rotation of the cut noncircular gear). The greatest difficulty, however, in cutting noncircular gears by this method is presented by the fact that each tooth space has a different profile and must be cut with a different cutter.

Noncircular gears can be cut by the generating method using standard gear-cutting tools (rack-type or rotary shaping cutters and gear hobs).

In producing the tooth profile of a noncircular gear by generation (Fig. 67*d*), it is necessary that the pitch curve rolls without slipping along the pitch line of the basic rack. This roll motion is accomplished by means of three interrelated motions $R_1T_2T_3$, one of these being a uniform motion and the other two nonuniform. The velocity ratios of these motions are variable.

In the majority of cases, noncircular gears are hobbed. A gear hobber of this type is considered in the following.

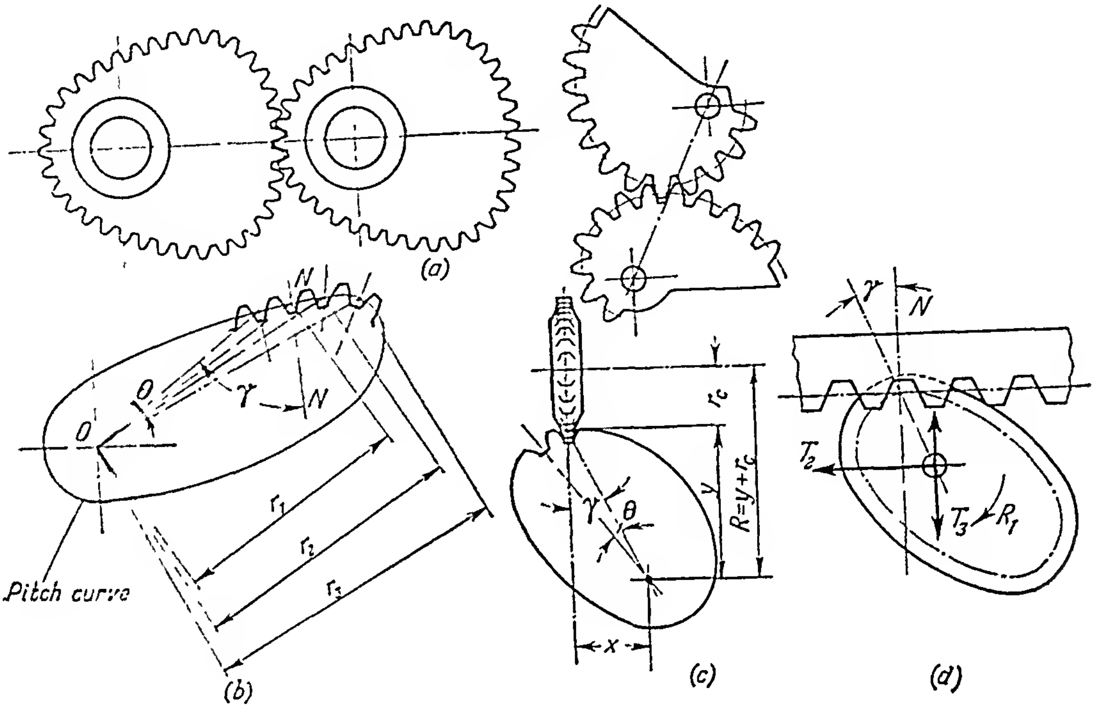

Fig. 67. Noncircular toothed gears

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In the majority of cases, noncircular gears are hobbed. A gear hobber of this type is considered in the following.

Noncircular Gear Hobbing Machine, Model E3-35

This hobber (Fig 68) can cut oval gears with a ratio of the major to the minor radii of the oval not exceeding 1 6 1 (8 5), maximum radius of the oval up to 100 mm and a module up to 2 5 mm using ordinary hobs, but of greater length (up to 140 mm)

Two formative motions are produced by the hobber (Fig 68*a*) the cutting motion F_v $(R_1 R_2 T_3)$ and feed motion F_s (T_4)

In the cutting motion, gear blank rotation R_2 is nonuniform

The kinematic group of the cutting motion is complex Its internal constraint consists of three internal kinematic chains, since the nonuniform rotation of the gear blank (Fig 68*b*) is composed of uniform rotation R_2 obtained as a result of uniform rotation of the table worm, and nonuniform rotation R_2'' produced by axial motion of the same table worm This motion is transmitted to the worm by interchangeable cam k_1 Thus, the internal constraint of the group for motion F_v $(R_1 R_2 R_2'' T_3)$ consists of the following internal kinematic chains

$R_1 \rightarrow \frac{60}{15} \times \frac{22}{22} \times \frac{21}{21} \times \frac{20}{24} \times \frac{e}{f} \rightarrow$ change gears $i_x \rightarrow$ gearing $\frac{36}{36} \times \frac{50}{50} \rightarrow$ worm gearing $\frac{1}{80} \rightarrow R_2$,

$R_2 \rightarrow$ worm gearing $\frac{80}{1} \rightarrow$ gearing $\frac{50}{50} \times \frac{45}{45} \rightarrow$ worm gearing $\frac{1}{40} \rightarrow$ cam $k_1 \rightarrow$ lever $\rightarrow$ narrow helical gear $50T \rightarrow$ axial motion of the table worm $\rightarrow$ $\rightarrow$ table $\overset{\neq}{R_2''}$ ($\neq$ is the conventional sign for nonuniform motion),

$R_2 \rightarrow$ worm gearing $\frac{80}{1} \rightarrow$ gearing $\frac{50}{50} \times \frac{45}{45} \rightarrow$ worm gearing $\frac{1}{40} \rightarrow$ cam $k_2 \rightarrow$ table saddle $\rightarrow \overset{\neq}{T_3}$

The helical gearing $\frac{50}{50}$ of the table is a differential because the narrow helical gear $50T$ has two degrees of freedom rotation about its axis and axial motion transmitted from cam k_1

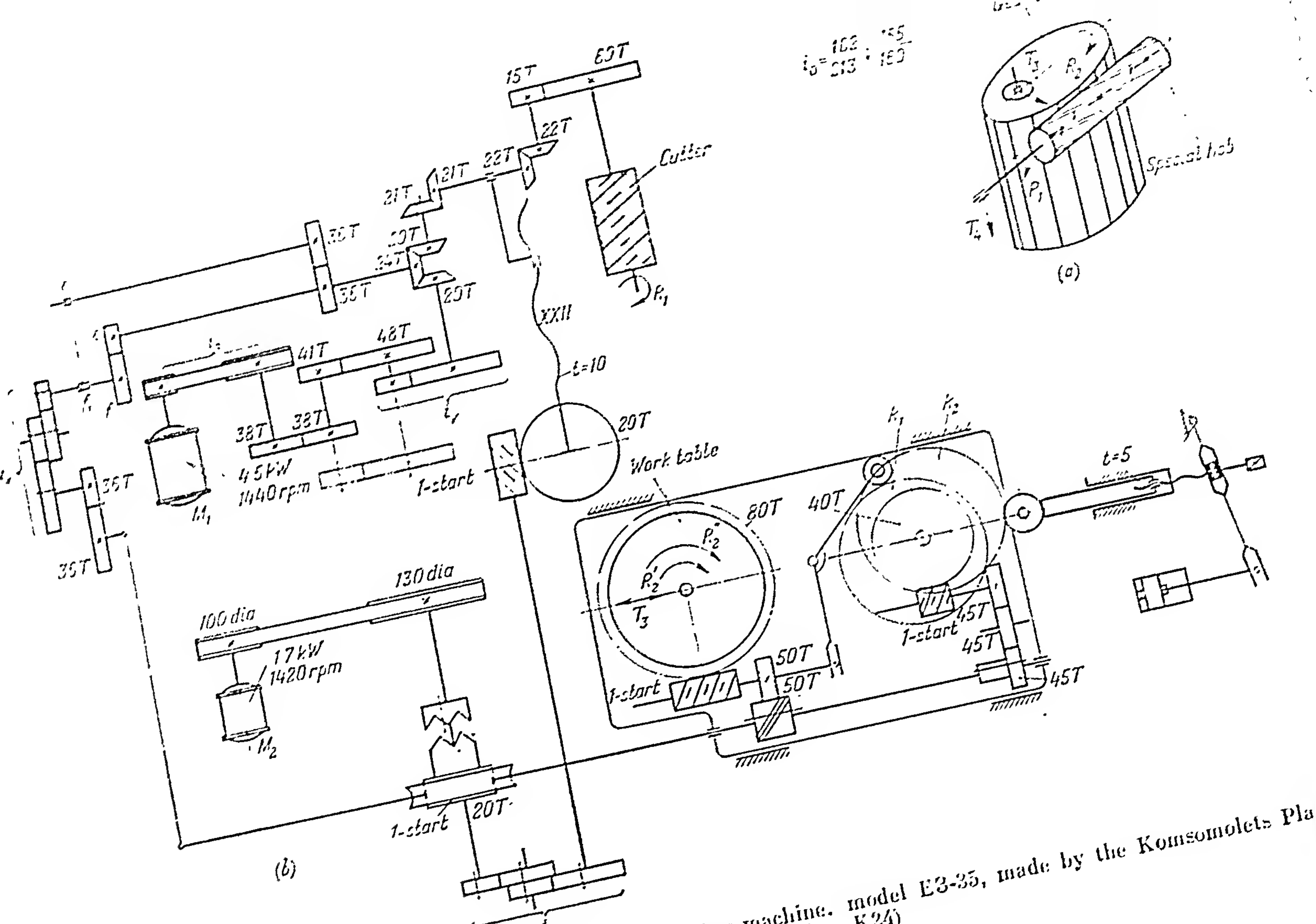
The external constraint of the cutting motion group is very simple It is connected to the internal constraint through bevel gear $24T$ The longitudinal feed group is also simple Change gears i_x and i_s are to be set up

Indexing change gears i_x

$$1 \text{ revolution of hob} \times \frac{60}{15} \times \frac{22}{22} \times \frac{21}{21} \times \frac{20}{24} \times \frac{e}{f} \times i_x \times \frac{36}{36} \times \frac{50}{50} \times \frac{1}{80} = \frac{k}{z}$$

hence

$$i_x = 24 \times \frac{f}{e} \times \frac{k}{z} \qquad (62)$$



...n of the noncircular-gear hobbing machine, model E3-35, made by the Komsomolets Plan[t]
Egoryevsk (structure class K24)

Longitudinal feed change gears i_s

$$1 \text{ revolution of work table} \times \frac{80}{1} \times \frac{50}{50} \times \frac{1}{20} \times i_s \times \frac{1}{20} \times 10 = s_v$$

and

$$i_s = \frac{s}{2} \tag{63}$$

Interchangeable cams k_1 and k_2 are manufactured separately for each size and type of gear to be cut

Oval helical gears can be hobbed in this machine using a nondifferential setup

Rapid traverse of the table saddle is available, it is driven from a separate motor M_2

Kinematic Structure of Machines for Cutting Threads with Nonuniform Pitch

Power screws with nonuniform pitch are used in machinery for the textile, paper and other industries Such screws are cut by special machine tools in which the operative motion of the cutting tool is along a helix of nonuniform pitch This type of thread cannot be specified merely by the lead P of the helix, which is constant in ordinary thread In this case, it is necessary to specify P_0 — lead of the first turn, P_z — lead of the last turn, z — number of turns in the given length of the thread S, $P \neq$ const—variable lead (or pitch) of the thread helix, φ—angle of spindle rotation, and S—length of tool travel (thread length)

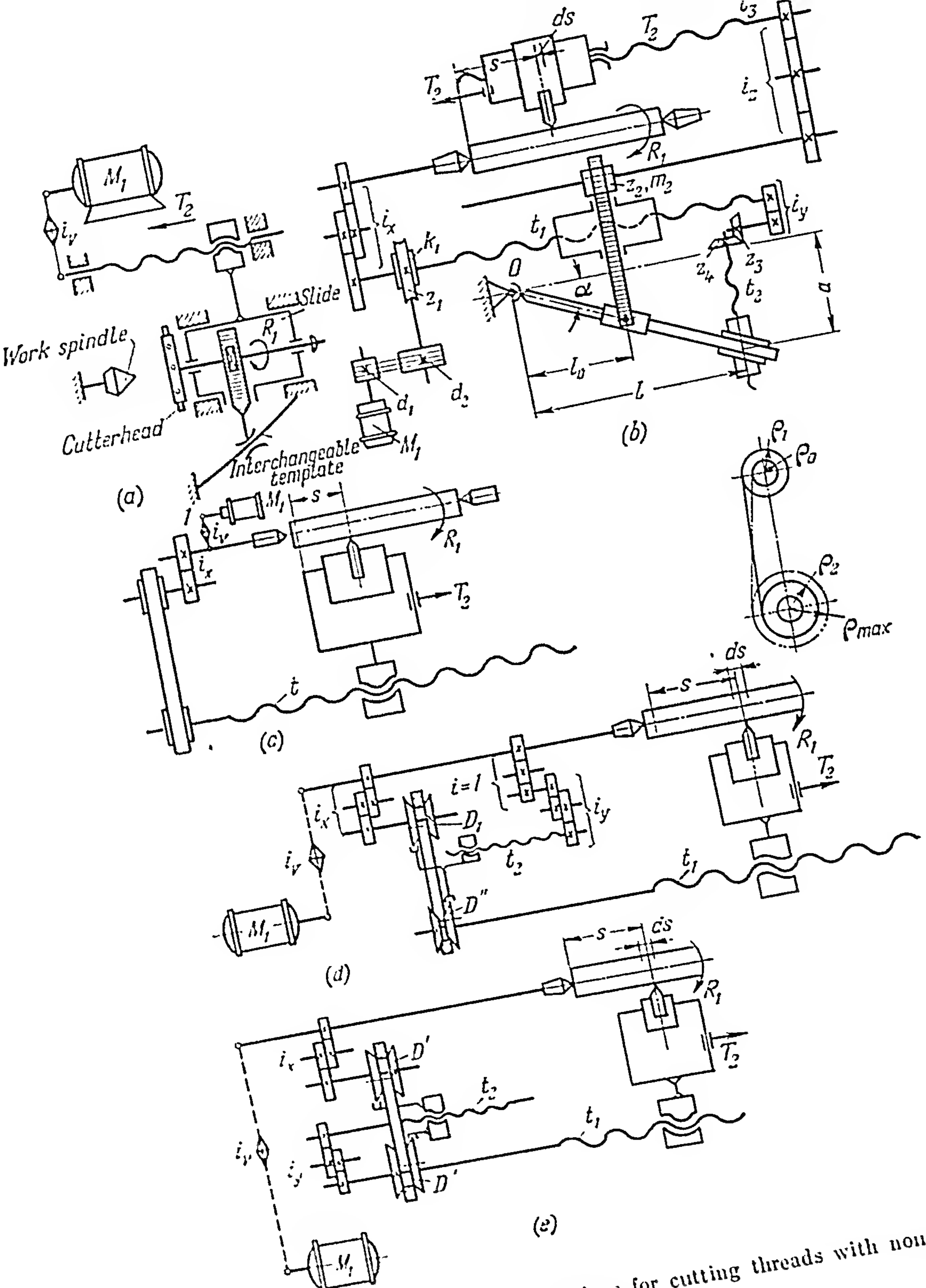
Several different models of machine tools are used in practice for cutting threads with nonuniform pitch

Illustrated in Fig 69*a* is the structure of a machine for cutting helical grooves with nonuniform pitch in the barrels of guns and rifles with a cutterhead having form tools Nonuniform rotation R_1 is produced in this machine by a curvilinear template Thread can be cut with pitch variation in a wide range according to any given law Only interchangeable templates with a curved profile of various curvatures are required for this purpose

The chief drawback in the operation of these machines is the difficulty in manufacturing sufficiently accurate wear resistant curvilinear templates

The machine shown schematically in Fig 69*b* also has a template in its internal constraint by means of which the helical motion F_v (R_1T_2) is produced with nonuniform longitudinal travel T_2 of the threading tool The template in this case, however is rectilinear and not interchangeable During operation this template or rather bar, swivels uniformly about pivot O

The machine is set up to the specified thread parameters by means of change gear units i_x, i_y and i_z These change gears only provide for the



... Kinematic structures of machines for cutting threads with nonuniform

quantitative aspects of the specified thread parameters They have no effect on the law according to which the thread pitch varies This law, in each case, follows from the nature of the nonuniformity mechanism itself A swivelling bar provides a parabolic development of the helix

The nonuniformity mechanism used in machines of the third type* consists of a drive with pulleys and a flexible steel band (Fig 69*c*) Upon uniform rotation of the driving pulley, the steel band is wound from the lower to the upper pulley Due to the change in the radii of the coils being wound and unwound, nonuniform motion is transmitted to the driven pulley and, with it, to the lead screw Here, the law governing the variation in nonuniform pitch is such as provided by a drive with a winding band, no other law can be obtained by this mechanism

The main shortcoming of a machine of this type is that only thread with a small difference in adjacent pitches can be cut

Illustrated in Fig 69*d* and *e* are the structural arrangements of machines in which the thread pitch is varied by means of conical pulleys and a V belt or steel ring ** In the first machine (Fig 69*d*), the conical half pulleys are spread and moved toward each other by a power screw with pitch t_2 which receives uniform motion from the machine spindle through change gears i_y In the second machine (Fig 69*e*), nonuniform rotation is transmitted to screw t_2 from the lead screw t_1 The laws of pitch variation differ somewhat in these two cases but their characteristics depend on the actual mechanism which produces the nonuniform motion

None of these machine tools have found widespread application and are, as a rule, pilot models developed in various engineering plants, since almost each new type of thread requires the development of a special new machine Special procedures may be required in each case to set up the change gears and other units

As an example, we shall consider the setting up procedure for one of these machines (Fig 69*b*) ***

We can write the kinematic balance equation for the internal constraint of the machine proceeding from the fact that during n revolutions of the spindle, the carriage with the cutting tool travels a distance of S mm

$$\left(n i_x t_1 \frac{a}{l} + n i_x i_y \times \frac{z_3}{z_4} t_2 \frac{l_0}{l}\right) \frac{1}{\pi m_2 z_2} \times i_2 i_3 = S \tag{64}$$

where $a = n\, i_x\, i_y \times \frac{z_3}{z_4} \times t_2$ = slide block travel in the cross direction

m_2 and z_2 = module and number of teeth of the rack pinion

* Designed by Z Platonov and P Kasonov

** This machine was developed in the Vtoraya Pyatiletka Plant

*** This setting up procedure was worked out by I M Kucher Cand Sc (Eng)

l and l_0 = slide block travel values in the longitudinal direction.

ubstituting the value of a in the equation we obtain

$$n i_x t_1 n i_x i_y \frac{z_3}{z_4} t_2 \times \frac{l_0}{l} \times \frac{1}{\pi m_2 z_2} i_z t_3 = S \tag{65}$$

$$n i_x t_1 n i_x i_y \frac{z_3}{z_4} t_2 \frac{1}{l} \frac{1}{\pi m_2 z_2} i_z t_3 + n i_x i_y \frac{z_3}{z_4} t_2 \frac{l_0}{l} \frac{1}{\pi m_2 z_2} i_z t_3 = S \tag{66}$$

nd finally

$$n^2 i_x^2 i_y \frac{z_3}{z_4} t_1 t_2 \frac{1}{\pi l m_2 z_2} i_z t_3 + n i_x i_y \frac{z_3}{z_4} t_2 \frac{l_0}{l} \frac{1}{\pi m_2 z_2} i_z t_3 = S$$

Since $\frac{dS}{dn} = P$, after differentiating the last equation we obtain

$$2 n i_x^2 i_y i_z \times \frac{z_3}{z_4} t_1 t_2 t_3 \frac{1}{\pi l m_2 z_2} + i_x i_y i_z \times \frac{z_3}{z_4} t_2 t_3 \times \frac{1}{\pi m_2 z_2} \times \frac{l_0}{l} = P \tag{67}$$

At $n = 0$, pitch P becomes the initial pitch P_0. Thus

$$P_0 = i_x i_y i_z \frac{t_2 t_3 z_3}{\pi l m_2 z_2 z_4} l_0$$

The pitch of the last turn of thread is determined from equation (67) at $n = z$. Hence

$$P_z = P_0 + 2 i_x^2 i_y i_z \frac{t_1 t_2 t_3 z_3}{\pi l m_2 z_2 z_4} z \tag{68}$$

The total length of the thread is determined from equation (66) by substituting $n = z$. Then

$$S = P_0 z + 2 i_x^2 i_y i_z \frac{t_1 t_2 t_3 z_3}{\pi l m_2 z_2 z_4} z^2 \tag{69}$$

Change gear ratio i_z can be determined from the following equation:

$$\frac{a_{max}}{\pi m_2 z_2} i_z t_3 = S \tag{70}$$

hence

$$i_z = \frac{S \pi m_2 z_2}{a_{max} t_3}$$

It is evident from the last equation that various lengths of thread are obtained by setting up change gears i_z. The value a_{max} is determined from the machine.

Still another specific equation can be written; it is evident that

$$z i_x i_y \frac{z_3}{z_4} t_2 = a_{max}$$

therefore

$$i_x i_y = \frac{a_{max}}{z t_2} \times \frac{z_4}{z_3}$$

From the equation $i_x^2 i_y i_z = \frac{P_z - P_0}{z} \times \frac{\pi m_2 z_2 l z_4}{2 t_1 t_2 t_3 z_3}$ we can determine

$$i_x = \frac{(P_z - P_0)}{2 t_1 S} l \tag{73}$$

and solve equation $i_x i_y = \frac{a_{max} z_4}{z t_2 z_3}$ for i_y; thus

$$i_y = \frac{2 a_{max} S}{z} \times \frac{z_4}{z_3} \times \frac{t_1}{t_2 (P_z - P_0) l} \tag{74}$$

Change gears i_x are set up to the value of the initial thread pitch P_0, and change gears i_y to the final thread pitch P_z.

The swivelling bar (template) with the slide block constitutes a differential since the slide block has two degrees of freedom, rectilinear travel along the bar and rotation about pivot O.

If a variable-pitch mechanism is used, other than the swivelling bar, the above setting-up procedure should be suitably revised.

l and l_0 = slide block travel values in the longitudinal direction.

Substituting the value of a in the equation we obtain

$$ni_xt_1ni_xi_y\frac{z_3}{z_4}t_2\frac{1}{l}\frac{1}{\pi m_2z_2}i_zt_3+ni_xi_y\frac{z_3}{z_4}t_2\times\frac{l_0}{l}\times\frac{1}{\pi m_2z_2}i_zt_3=S \tag{65}$$

and finally

$$n^2i_x^2i_y\frac{z_3}{z_4}t_1t_2\frac{1}{\pi lm_2z_2}i_zt_3+ni_xi_y\frac{z_3}{z_4}t_2\frac{l_0}{l}\frac{1}{\pi m_2z_2}i_zt_3=S \tag{66}$$

Since $\frac{dS}{dn}=P$, after differentiating the last equation we obtain

$$2ni_x^2i_yi_z\times\frac{z_3}{z_4}t_1t_2t_3\frac{1}{\pi lm_2z_2}+i_xi_yi_z\times\frac{z_3}{z_4}t_2t_3\times\frac{1}{\pi m_2z_2}\times\frac{l_0}{l}=P \tag{67}$$

At $n=0$, pitch P becomes the initial pitch P_0. Thus

$$P_0=i_xi_yi_z\frac{t_2t_3z_3}{\pi lm_2z_2z_4}l_0$$

The pitch of the last turn of thread is determined from equation (67) at $n = z$. Hence

$$P_z=P_0+2i_x^2i_yi_z\frac{t_1t_2t_3z_3}{\pi lm_2z_2z_4}z \tag{68}$$

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Change gear ratio i_z can be determined from the following equation:

$$\frac{a_{max}}{\pi m_2z_2}i_zt_3=S$$

hence

$$i_z=\frac{S\pi m_2z_2}{a_{max}t_3} \tag{70}$$

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Still another specific equation can be written; it is evident that

$$zi_xi_y\frac{z_3}{z_4}t_2=a_{max} \tag{71}$$

therefore

$$i_xi_y=\frac{a_{max}}{zt_2}\times\frac{z_4}{z_3} \tag{72}$$

From the equation $i_x^2 i_y i_z = \frac{P_z - P_0}{z} \times \frac{\pi m_2 z_2 l z_4}{2 i_1 i_2 i_3 z_3}$ we can determine

$$i_x = \frac{(P_z - P_0)}{2 i_1 S} l \tag{73}$$

and solve equation $i_x i_y = \frac{a_{max} z_4}{z i_2 z_3}$ for i_y, thus

$$i_y = \frac{2 a_{max} S}{z} \times \frac{z_4}{z_3} \times \frac{i_1}{i_2 (P_z - P_0) l} \tag{74}$$

Change gears i_x are set up to the value of the initial thread pitch P_0, and change gears i_y to the final thread pitch P_z

The swivelling bar (template) with the slide block constitutes a differential since the slide block has two degrees of freedom rectilinear travel along the bar and rotation about pivot O

If a variable pitch mechanism is used, other than the swivelling bar, the above setting up procedure should be suitably revised

CHAPTER 6

KINEMATIC STRUCTURES OF MACHINE TOOLS HAVING NONMECHANICAL KINEMATIC CONSTRAINTS

Mechanical and Nonmechanical Constraints

Kinematic constraints based on mechanical members are extensively employed in drive trains and internal kinematic chains. If internal trains of mechanical members are of excessive length, as for example, in heavy machine tools, they become unwieldy and consequently cannot always ensure sufficient kinematic accuracy in the operation of the train. If, in addition, these trains operate under severe dynamic conditions, transmitting large forces to the final members, the elements of the trains wear rapidly and the initial accuracy of the machine tool is soon lost. Such trains do not always operate properly in high-speed and precision machine tools as well. Attempts have been made in recent years to devise the required internal kinematic constraints using kinematic chains of nonmechanical elements, based on energy transmission by electrical, hydraulic or other means.

Notwithstanding their advantages over mechanical constraints (longer kinematic chains can be employed, their members are less subject to wear, and the initial kinematic accuracy is retained over a longer period of operation), electrical, hydraulic and other types of nonmechanical constraints have not yet found widespread application in machine tools, especially of the general-purpose types. This is accounted for by the fact that these constraints do not possess the kinematic stability of trains with mechanical members. Due to various inherent reasons, electrical or hydraulic constraints are easily and frequently impaired; they cannot always maintain constant ratios between the kinematic parameters of the motion of the final members and, consequently, the required kinematic accuracy.

Stable operation of such constraints sometimes depends upon conditions that are difficult to comply with during regular operation of the machine tool. For example, the accuracy of operation of a hydraulic constraint depends upon the temperature of the working fluid, resistances in the hydraulic system, leakage, etc.

These circumstances lead to the inevitable use of various supplementary devices for creating and maintaining conditions vital for normal operation of these nonmechanical kinematic constraints. Such devices naturally complicate the construction of the machine tool. Hence, it is not always expedient to employ electrical or hydraulic devices as internal constraints,

at any rate until more highly perfected elements of such systems have been developed.

At the present time, simpler versions of electrical and hydraulic systems are used as intergroup connections providing synchronous operation of various units of a machine tool. They are applied, for example, to synchronize the feed motions of the two heads of milling and centring machines, and to obtain a definite feed per revolution in machine tools having drive motors with different power characteristics, in particular, in machines having an electric motor in the cutting motion group and a hydraulic motor in the feed motion group.

As internal constraints within a single kinematic group, nonmechanical constraints are used to a fairly wide extent in tracer-controlled machine tools, including milling machines of this type.

For all practical purposes, the question of employing electrical or hydraulic constraints in machine tools is solved, in most cases, by the application *of combination constraints, such as electromechanical, hydromechanical,* hydroelectrical, etc., utilizing the specific advantages of each type.

As more advanced electrical, hydraulic, pneumatic and other allied devices become available, the application of electrical, hydraulic and other nonmechanical constraints shows much promise because their use greatly simplifies the construction of machine tools.

Several machine tools with nonmechanical constraints shall be taken up in the following.

Gear Grinder of the Synchrotic Type

This machine (Fig. 70) can grind circular spur and helical gears of a diameter from 12 to 240 mm and with a module from 0.5 to 4 mm using a helically profiled grinding wheel having a horizontal axis of rotation.

The method of generating the tooth profile is the same as in the model 5A833 gear grinder (see Fig. 55), and therefore the cutting motion is $F_v (R_1R_2)$.

The method of producing the helix along the tooth length is the same as in a gear hobber, and the feed motion is $F_s (T_3R_4)$, except that the elementary *motion* T_3 *along an element* of the cylinder (Fig. 70) is imparted to the work gear and not to the tool (wheel).

To develop the side surfaces of the worm thread on the wheel with a diamond or crusher roll, the helical operative motion $F_t (T_5R_6)$ is required in which T_5 is longitudinal travel of the diamond and R_6 is rotation of the grinding wheel. Thus three complex operative motions F_v, F_s and F_t are produced by the grinder, and the kinematic structure consists of three separate kinematic groups (not including the controlling motion group).

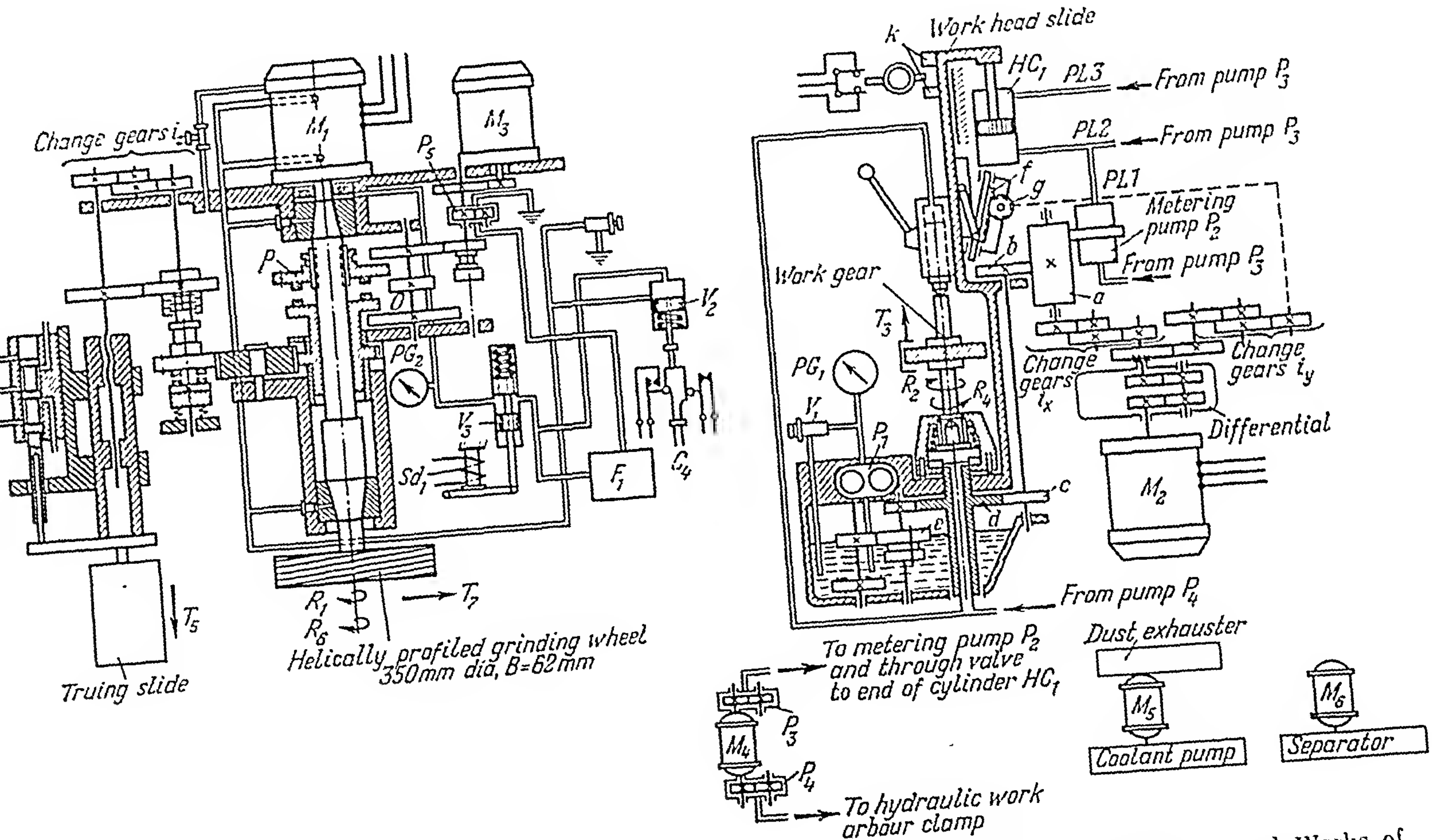

Fig. 70. Kinematic diagram of the synchrotic type gear grinder, model ZA, made by the Reishauer Tool Works of Switzerland (structure class K24)

Like any other kinematic group, the kinematic group of the cutting motion consists of an internal constraint and the drive This internal constraint connects the elementary motions R_1 and R_2, i e it is between the spindle of the helically profiled wheel and the work spindle The internal constraint comprises a single kinematic chain with several unusual members (links) It has the following structure wheel spindle → electric motor M_1 → → electric circuit → electric motor M_2 → differential → change gears i_x → → gears $\frac{a}{b} \times \frac{c}{d}$ → the work spindle The constraint governing the path of the generating motion $F_v (R_1R_2)$ must operate with great accuracy but it has no precision worm gearing units at the work and wheel spindles as in other machines Along with the rigid, or positive, members, such as toothed gears, this constraint has elements (electr ... ric circuit) whic' ... cs Despite this ... ng ensured in th ... quir rel cage electr ... a sufficiently flat characteristic, and can operate synchronously when on steady duty and not subject to especially sharp fluctuations in power supply voltage (not exceeding ±5 per cent) In starting the grinder, electric motor M_2 is started by direct connection with the power supply, while electric motor M_1 is started through a star delta relay and a time delay relay The motor is automatically switched over to a delta connection at the end of 4 seconds

To maintain a more or less constant contact pressure between the wheel and work gear so as to oppose variations in load in grinding, a braking load pump BP_1 is installed in the kinematic chain between electric motor M_2 and the work spindle This pump is driven by the work spindle through a sliding double cluster gear e Depending upon the number of teeth and helix angle of the gear being ground the required operating conditions of pump BP_1 are provided by means of cluster gear e and valve V_1 which is adjusted to pressure gauge PG_1 For example, in grinding helical gears the pump must operate at a higher pressure than when spur gears are ground Change gears i_x of this kinematic chain enable it to be set up in respect to the path of the motion

The internal kinematic chain is powered by electric motors M_1 and M_2 which are themselves included in the chain There is no need for setting up this grinder to the cutting (grinding) speed, and the motors have strictly constant speeds

The structure of the second kinematic group for the feed motion F_s is also somewhat unusual The internal constraint between the elementary motions T_3 and R_4 is provided between the work head slide and work spindle by means of a swivelling bar with rack f rack pinion g, change gears i_y, housing (planet carrier) of the differential, change gears i_x and the gears

CHAPTER 7

EFFECT OF VARIOUS FACTORS ON THE KINEMATIC STRUCTURE OF MACHINE TOOLS

The typical kinematic structure of any machine tool complies in the main with requirements of a kinematic nature, and does not always fully meet other service requirements, especially those concerned with accuracy and production capacity. These requirements are usually met, for the most part, by a suitable construction. In certain cases, however, better results can be attained when definite modifications are made in the typical kinematic structure of the machine tool.

Let us consider these possibilities for changing the typical kinematic structure in accordance with the processing purpose of the machine tool, its universality, the level of the requirements made to the machining accuracy and output, dynamic factors associated with operation of the machine tool, and requirements made to the setting-up facilities.

Depending upon the processing requirements, machine tools of the same processing group may have different kinematic structures. The group of boring machines can be cited as an example. If in boring a workpiece, end flanges must be faced in the same setting, a facing slide is provided on the faceplate. This slide has cross feed and carries a single-point tool called, in this case, a fly cutter. The kinematic chain that powers tool cross feed (Fig. 71*a*) includes a hidden summation drive resulting from gear *c* rolling about gear *b* since the slide is mounted on the rotating faceplate. This summation drive imparts supplementary radial motion to the facing slide and the fly cutter so that the actual radial feed does not comply with the normal expected rate. To compensate for this additional displacement, a differential (Fig. 71*b*) is built into the feed group. Two motions are transmitted through the differential: one is a slow radial feed motion through change gears i_s while the other is a rapid motion, along gear train $1\text{-}4\text{-}5\text{-}6\text{-}\frac{a}{b}$, for compensating for the surplus radial displacement due to the hidden planetary transmission. Thus the kinematic structure of the boring machine has become more complex with the application of a fly cutter.

In addition to the ordinary elements and mechanisms required to bore holes, the design of a jig borer incorporates very complex devices for reading off the co-ordinate dimensions. These devices are required to enable the

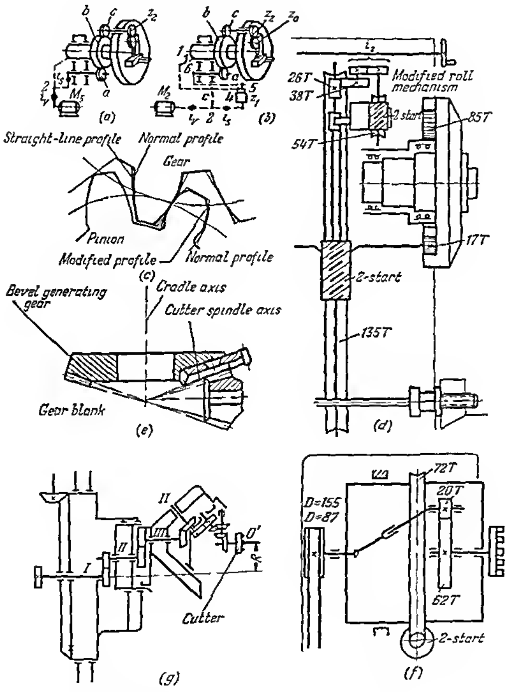

Fig 71 Modifications of the typical structure of various machine tools

holes being bored to be positioned at the specified co-ordinates, and greatly alter and complicate the kinematics of an ordinary boring machine.

Still another example can be mentioned in which the typical kinematic structure is modified to meet the processing requirements.

If a spiral bevel gear generator is to be used to cut Formate gears (Fig. 71*c*), in addition to the conventional spiral bevel gears, then the typical kinematic structure must be changed. The changes that are to be made depend upon which of the two methods for cutting Formate gears is used. When the first method is applied, the cradle has a nonuniform motion for cutting the pinion member (smaller of the pair), such motion being provided by the modified roll mechanism built into the generator (Fig. 71*d*). This nonuniform rotation of the cradle is obtained as a result of two motions of the cradle worm: uniform rotation and nonuniform axial motion transmitted by the modified roll mechanism through its change gears i_z. The second method of cutting a Formate pinion involves the setting of the face-mill type cutter at an angle to the cradle axis (Fig. 71*e*). In this case, the pinions are cut by means of a bevel generating gear and not a crown gear as ordinarily. Motion can be transmitted to the inclined cutter through shafts that are not in alignment, either by means of a telescopic shaft with universal joints (Fig. 71*f*) which are not capable of transmitting heavy torques or, through further complication of the drive, enabling motion to be transmitted by ordinary gearing between nonaligned shafts (Fig. 71*g*). A drive of the second type can easily accommodate heavy loads. In either case, the kinematics of the generator, as far as the drive to the cutter is concerned, differs from the typical structure of a spiral bevel gear generator. Many other examples could be cited showing the further development of the kinematic structure when more extensive demands are made to the processing capacity of a machine tool. This tendency is quite clear however from the preceding examples.

The kinematic structure of a machine tool may be expanded and made more complex in cases when it is necessary either to extend the processing capacities or to increase the output by increasing the number of spindles, clamping stations or sections, or when the machine is to be built into an automated production line or transfer machine.

When the processing capacities of a machine tool are extended, it becomes a multiple-structure machine in which several different combinations of formative, indexing and feed-in operative motions can be obtained. This enables work of various shapes to be machined by various cutting tools in performing different processing operations. By and large, it can be said that the kinematic structure of an omniversal machine tool allows for the performance of all the required operations. If only one operation is being performed, only a part of the machine with its partial structure is being utilized. Hence, provision must be made for devices that enable the machine

to be changed over from one partial structure to another Various procedures are employed in such multiple-structure machine tools to obtain several different partial structures

There are three main procedures for changing over from one partial structure to another The first of these involves the use of the available kinematic operative members one after another without removing any from the machine which, *consequently*, does *not have a single interchangeable unit* The second procedure consists in changing, not only various members but whole units of the machine Both of the preceding methods are employed in the third procedure, in which interchangeable units are utilized only partially

An example of a multiple structure machine tool is one for producing globoid worm gearing Model 549, for instance consists of twelve partial structures It can be used to machine globoid worms and their conjugate worm wheels by means of cutter heads, hobs, shaving cutters, grinding wheels and laps, and to perform both roughing and finishing operations This machine has various special devices by means of which any particular partial structure may be set up for operation Such devices include for example a number of jaw clutches In addition to these interstructure devices the machine has three hobbing and grinding attachments which are used in conjunction with one of the available partial structures

Machine tool structures may sometimes differ to some extent if the machine tools are of the multiple spindle or multiple-station type in which several workpieces are machined simultaneously Here new blanks are loaded and the finished work is removed simultaneously with the machining of other blanks In these machines the kinematic structure is made more complex by the use of parallel kinematic chains, so that a multiple-station machine is obtained

Parallel stations of a machine several spindles or slides are sometimes designed in the form of separate sets or units having separate, independent but repeated kinematic structures in which case the machine tool is of the multiple-section type As a rule, all the sections are mounted on a common bed, or base Machine tools of this type can handle identical or different parts simultaneously An example is the two section gear hobbing machine having two sections identical in kinematics and construction mounted on a common base

The general kinematic structure of all multiple spindle multiple-station and multiple-section machine tools is complicated by the provision of supplementary identical kinematic groups and intergroup kinematic constraints

Machine tools based on a diversified structure i e multiple structure types cannot be placed into any definite class of typical kinematic structures Therefore the general structure of such machines is determined by

the class of the partial structure which produces formative operative motions composed of the maximum number of elementary motions.

When a machine tool is built into a transfer machine or automated production line, the kinematic groups for the controlling motions are considerably expanded and become much more complex. As a result, the kinematic structure of each separate machine tool may also be changed.

It is hardly expedient to take up the kinematic structure of automatic transfer machines in the present part of this textbook. Nevertheless, the theoretical principles formulated above are applicable in analysing the kinematic structure of a transfer machine if it is considered as a whole, i.e. as a separate unit. The kinematic structure of any component machine tool, then, consists of intragroup and intergroup kinematic constraints, which are to be regarded as intramachine constraints in relation to the whole transfer machine. Consequently, a transfer machine, or automated production line, can be considered to be made up of intramachine and intermachine constraints. The latter possess many specific features, and will be taken up in Part Six of Vol. 4.

Higher machining-accuracy requirements may be met to some extent by changing the typical kinematic structure along the following lines: improving the structure of the internal kinematic constraints; increasing the number of elementary motions making up complex operative motions, correction devices being used for this purpose; and by applying nonmechanical internal constraints. The accuracy of internal kinematic chains can be increased by using spur gears in them instead of helical or bevel gears, or other types of gearing on intersecting or crossed axes. This principle can be demonstrated by the example of the gear-shaping machines, models 5A12 and 5B12.

These two models are very similar in their capacities, quantity and type of kinematic groups, arrangement and construction features. They differ in the number and type of members that compose the internal kinematic indexing chain with change gears i_x (Fig. 72), interconnecting shaping cutter rotation with blank rotation. Both models have precision worm gearing units for driving the cutter ram and work spindle, but the train between the worm shafts of these drives includes four pairs of bevel gears and two other pairs of gears in model 5A12 (Fig. 72), while eleven pairs of spur gears are provided in this train of model 5B12 (Fig. 73). The second model has no bevel gearing.

The substituting of spur gears in the indexing train of model 5B12 for bevel gearing enables this gear shaper to produce gears of a higher accuracy class than those made by model 5A12.

However, this change in the structure of the indexing gear train has led to an increase in the number of gears (model 5A12 has 16 gears while model 5B12 has 21), but the accuracy of the indexing train has been substantially raised in model 5B12. The kinematic accuracy of the latter model

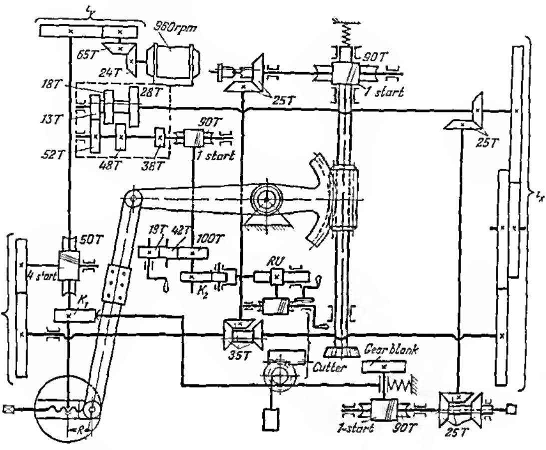

Fig 72 Kinematic diagram of the gear shaper model 5A12

could be further increased by reducing the number of gears and by excluding one reversing device, RU_1 or RU_2, from the gear train One reversing device, or unit (either RU_1 or RU_2) is required to set up the shaper for cutting internal gears the second enables the whole generating motion to be reversed The latter is employed in cases when the cutting edges on one side of all the cutter teeth have become dull while they are still sharp on the other side These sharp edges are utilized by reversing the generating motion, thereby considerably prolonging the life of expensive cutters. Since the second reversing unit must change the direction of the whole operative motion F_v $(R_1 R_2)$, it should be within the external kinematic constraint, i.e. in the feed train, and not in the internal constraint as it has been arranged in models 5A12 and 5B12

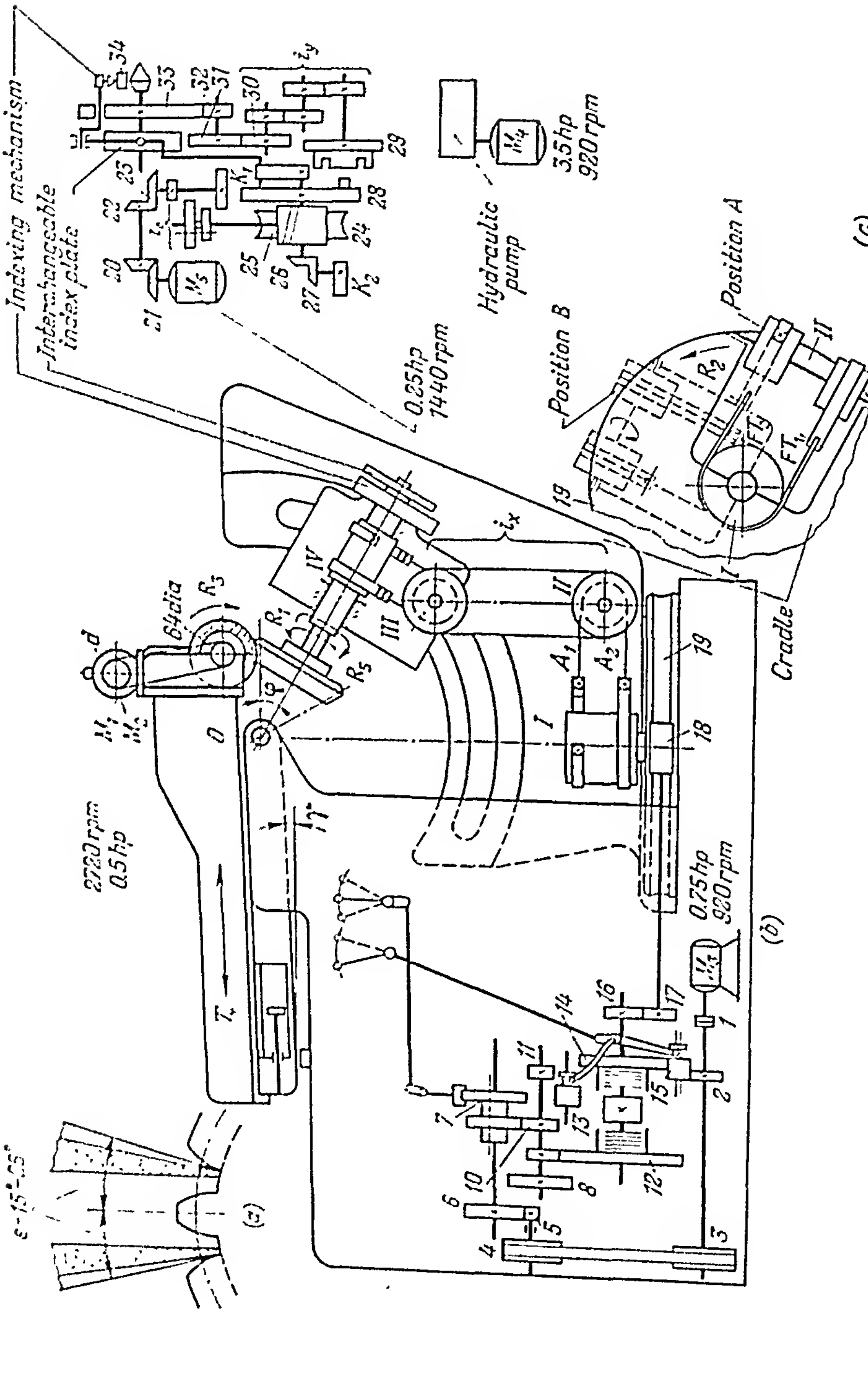

Fig. 74. Kinematic diagram of the gear grinder, model KS-42, made by the Maag-Gear Wheel Co. of Switzerland

A simpler kinematic structure can be obtained if each kinematic group of a machine tool has its own separate motor, and motors with different, more suitable, power characteristics can be employed. It is impossible, however, in this case to maintain a constant rate of feed per spindle revolution, which is of prime importance in finish machining. Constant feed per revolution can be obtained by building an intergroup constraint into the system. For instance, in the heavy-duty lathe, shown in Fig. 75*c*, the spindle is driven by an electric motor and the carriage is longitudinally traversed by a hydraulic cylinder. The intergroup constraint passes from the spindle through points *3* and *4* to the lead screw which is linked to valve V_1. The lead screw has two axial motions: T_2—in one direction from carriage travel and powered by the hydraulic cylinder, and T_1—in the other direction from rotation of the screw. These motions are always equal in velocity. If there is a change in velocity of either the motor or the cylinder piston, the intergroup constraint automatically shifts the piston of valve V_1, changing the velocity of the cylinder piston so as to maintain a constant rate of feed per revolution. The kinematic structure of a machine tool becomes more complex with the introduction of an intergroup constraint.

Any reduction in handling and setup time increases the output of a machine tool. In some cases, the kinematics may be made more complicated so as to reduce the time required to set up the kinematics and processing arrangements of a machine tool. For example, an extra differential gear train is incorporated in the design of a relieving machine.

Figure 42 showed four versions of the kinematic structure for relieving machines. The changes were made in the structure in these versions with the aim of simplifying the procedure for setting up the kinematics.

The indexing change gears i_x are arranged in gear hobbers (see Fig. 20*a*) so that in cutting the conjugate gear of a pair it is not necessary to change over the change gears i_y thus reducing the setup time.

The kinematic structure of a machine tool may be modified to comply with other factors and requirements not considered here. It is evident from the examples given above that these changes do not disarrange the typical structure, but only alter it to some extent and supplement it.

The possibilities of improving the kinematic structures of machine tools to increase their accuracy, production capacity and stability of performance are not, of course, confined to the examples given in this chapter.

PART FOUR

HYDRAULIC SYSTEMS OF METAL-CUTTING MACHINE TOOLS

A simpler kinematic structure can be obtained if each kinematic group of a machine tool has its own separate motor, and motors with different, more suitable, power characteristics can be employed. It is impossible, however, in this case to maintain a constant rate of feed per spindle revolution, which is of prime importance in finish machining. Constant feed per revolution can be obtained by building an intergroup constraint into the system. For instance, in the heavy-duty lathe, shown in Fig. 75*c*, the spindle is driven by an electric motor and the carriage is longitudinally traversed by a hydraulic cylinder. The intergroup constraint passes from the spindle through points *3* and *4* to the lead screw which is linked to valve V_1. The lead screw has two axial motions: T_2—in one direction from carriage travel and powered by the hydraulic cylinder, and T_1—in the other direction from rotation of the screw. These motions are always equal in velocity. If there is a change in velocity of either the motor or the cylinder piston, the intergroup constraint automatically shifts the piston of valve V_1, changing the velocity of the cylinder piston so as to maintain a constant rate of feed per revolution. The kinematic structure of a machine tool becomes more complex with the introduction of an intergroup constraint.

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PART FOUR

HYDRAULIC SYSTEMS OF METAL-CUTTING MACHINE TOOLS

CHAPTER 8

GENERAL PRINCIPLES OF HYDRAULIC DRIVES

8-1. Operation of a Hydraulic System and its Specific Features

ydraulic systems have found versatile application in machine tool neering They are employed, for instance, in unit built machine tools, sfer machines and programme controlled machines In machine tools ertain groups (broaching machines, grinders, etc) they have almost pletely superseded other types of drives At present, hydraulic systems being used in synchronizing devices, in travel control systems, etc re are many indications that the further automation of manufacturing esses in the engineering industries will undoubtedly develop along lines of a more extensive application of automatic hydraulic ces.

ydraulic systems were incorporated in the design of machine tools nning with about 1925 They were first applied in machine tools requir low pressure hydraulic fluid, for example, grinders Hydraulically ered machines operating at low speeds and high pressures were designed he ENIMS Institute in 1934 35 These were unit-built drilling, boring milling machines Later on, this institute did considerable work in the l of standardizing various hydraulic devices and hydraulic circuits for hine tools of many types As a result, much less time was required to achine tools and advanced designing tech use of unified control devices tested in the various plants

Iydraulic fluids can be used to effect any continuous or intermittent ions of power members, including rectilinear reciprocation and rotation ch are the most widely used in machines

Iydraulic systems, employed in up-to date machine tools to obtain both hese types of motion, operate on principles that were described to some nt in the preceeding chapters (see Secs 2 11, 4 5 and 8-4 of Vol 1)

ny type of hydraulic system consists of two main parts a *pump*, the nary part of the system, and the *hydraulic motor* the secondary part of system, supplied by the pump The pump develops the pressure or head he working fluid by expending mechanical energy A hydraulic motor of reciprocating type (*hydraulic cylinder*) or rotary type is intended for verting the head developed by the pump into mechanical work Thus the

head of the pump provides the required force in the hydraulic cylinder or the torque on the shaft of the rotary hydraulic motor, and the fluid flow provides the required velocity of the working member. Hence, a hydraulic system can be regarded as a converting machine (power transformer) with a closed power cycle.

The pump and hydraulic motor are connected with the controls which provide for the required sequence of all the stages of the working cycle of the machine tool. The controls include devices for varying speeds and for reversing and reducing and check (nonreturn) valves, relays, etc. The controls may be mounted on a special *panel*.

Energy losses in a hydraulic system are made up of: (a) *volumetric losses*, due to leakage of the working fluid; (b) *hydraulic losses*, due to a drop in pressure, and (c) *mechanical losses*, due to friction.

The speed of a hydraulic motor (reciprocating or rotary) is varied by regulating the amount of fluid passing into or out of the motor in unit time; motors are reversed by changing the direction of flow of the fluid delivered to the motor. Uniform motion of a hydraulic motor and, consequently, constant speed of units rigidly linked to the motor, are possible if the load on the motor due to all kinds of resistance remains constant, if the hydraulic system is leak-proof and contains no air.

In conversion machines with a closed power process, additional power losses, associated with the conversion of one form of energy into another, are inevitable and are characterized by the efficiency value.

Hydraulic drives with power rating of $N \geqslant 0.45$ kW are economically sound; electrical drives are usually used for $N < 0.45$ kW.

A hydraulic drive is advisable for developing high torques and pulling forces. The cost of designing, manufacture, installation and operation of such a drive is substantially less than for an electrical drive of the same power rating. The initial cost of the hydraulic equipment is from 60 to 67 per cent lower.

The speed of response of a system and its dynamic properties are evaluated, for practical purposes, by the ratio of the maximum torque T_{max} (or maximum pulling force P) to the moment of inertia I of the rotor, i.e. by the value $a = \frac{T_{max}}{I}$. It is well known that this ratio is limited in electrical engineering by the permissible torque per unit weight of the iron in the rotor of the electric motor. The ratio a drops with an increase in power. The maximum torque developed by a hydraulic system is limited only by the strength of the component parts, hence, by the chosen materials.

The dynamic properties and high-speed response of a hydraulic drive being designed can be improved by reducing the mass of the moving parts. Modern hydraulic drives can attain a weight-to-power ratio of 0.55 to 0.8 kg per kW, values as yet unattainable in an electrical drive.

The overall dimensions of electrical equipment and electrical devices are determined by the temperature conditions and the *magnetic flux density*, this value does not exceed 0 2 0 25 T (tesla) for high quality electrical steel or permeodur alloy This corresponds to a specific torsional force of about 20 kgf per sq cm and the designer is again restricted in his choice of a suitable material The designer enjoys much more freedom in selecting materials for hydraulic devices

The operation of hydraulic systems in machinery in general, and especially in machine tools, has shown that the maximum length of a pressure line rarely exceeds 40 or 50 m On the contrary, the length of an electrical transmission line is not restricted

As concerns response accuracy, the possibilities of electrical and hydraulic devices are about the same A hydraulic drive, however, can have facilities for stepless variation of velocity and pressure in accordance with any specified law these facilities being much simpler than when electrical or electronic devices are used The accuracy of travel under load, as in hydraulic tracer controlled machines, averages about 0 01 mm

After it is installed, a hydraulic system is to be set up only in respect to a single parameter—pressure—while electrical systems and, *all* the more, electronic systems, must be set up in respect to several parameters Hence, it is considerably easier to locate possible defects in the assembly and installation of a hydraulic system

8-2. Hydraulic Fluids Used in Machine Tool Drives

Mineral, or petroleum, oils of various grades (indicated below) are used as the working fluid in the hydraulic drives of machine tools to transmit pressure and to actuate hydraulic motors Under severe operating conditions, for prolonged continuous duty, as well as under high or low temperature conditions hydraulic fluids with elevated oxidation stability and high viscosity are required Special additive treated mineral oils are employed in these cases

The main service property used in selecting and comparing oils is the *viscosity index* which shows the change in the viscosity of an oil with its temperature The higher the viscosity index the higher the quality of the oil and the better it has been refined A substantial increase in the viscosity index can be achieved by the application of special synthetic oil additives

Mineral oils have found predominant application in the hydraulic drives of metal cutting machine tools Experience shows that an oil viscosity index of about 90 is most suited for such hydraulic drives

High-viscosity oil is usually recommended for high operating pressures, when $p \geq 100$ kgf per sq cm, to reduce volumetric losses and to maintain a normal viscosity at an elevated temperature.

Radial rotary piston pumps, operating at pressures of $p > 175$ kgf per sq cm, perform satisfactorily if use is made of oils with a kinematic viscosity of 1 to 2 sq cm per sec at a temperature of $T = 40°C$. A viscosity of 0.35 to 0.65 sq cm per sec is recommended for pressures of $p < 70$ kgf per sq cm. Many foreign manufacturers recommend oil with a viscosity of 0.6 to 0.65 sq cm per sec for their hydraulic drives.

In the USSR, Industrial oil 12 or 20, according to USSR Std GOST 1707-51, is employed in translatory motion drives operating at medium pressures of $p = 30$ kgf per sq cm. Oils of higher viscosity are used in rotary motion drives. These include Turbine oil 22 according to USSR Std GOST 32-53, and Machine oils 30 and 45 according to GOST 1707-51.

The oil must be sufficiently homogeneous as to its chemical composition. This homogeneity is checked by determining the flash point. Other things being equal, the better an oil is refined, the more homogeneous it is. The freezing point of an oil is an indication of its moisture content. Moisture interacts with the metal oxides of various rubbing components in the pump, hydraulic motor, etc., and the hydrates of these oxides interact, in turn, with the organic acids to produce soap which serves as a catalyst in the oil oxidation processes and thus deteriorates the operating properties of the oil.

Condensate of atmospheric moisture accumulates quite rapidly in hydraulic oil. Most of the condensate settles to the bottom of the tank, or reservoir, but the remainder mixes with the oil, forming an emulsion which circulates in the hydraulic system. Therefore, it is necessary to clean the remains of the used oil from the reservoir before pouring in fresh oil.

Of especial importance is the chemical stability of an oil, which is characterized by its oxidability at high temperatures and when it is subject to the catalytic effect of metals. Oil oxidation is accompanied by the formation of resinous deposits that gum up the orifices of control devices.

The application of the antioxidant additive ДФ-1 inhibits or even stops the aging and oxidizing processes in oils.

Aeration causes oil to foam, creating conditions that favour the formation of air emulsions of very low density; the presence of such emulsions violates uniform motion of the travelling members in a machine tool. In most cases, silicon polymers (polymethylsiloxanes) are used as antifoam agents.

Used as hydraulic fluids in the USSR are oils refined from Soviet eastern paraffin-base or sulphur-bearing crudes, without additives, in the pure state. The viscosity index of such oils is double that of oils from low-sulphur Azerbaijan crudes.

On the basis of investigations, the Petroleum Research Institute of the USSR has worked out the formula of special oils for use in hydraulically

operated machine tools These oils have a kinematic viscosity from 0 3 to 0 6 sq cm per sec at 50°C and are based on well refined turbine oil from eastern crudes into which various additives have been introduced The grade designation is ВНИИ НП-403 and the viscosity index is 92МР according to Specifications ТУ 12Н No 6 6? As concerns certain of its features (for instance antifoam property) it is superior to many imported grades

The extensive application of hydraulics in machinery power systems and especially in tracer control systems has accented the importance of such oil properties as compressibility and ability to dissolve air Dissolved air violates the uniformity of hydraulic motor operation affects the oil viscosity and reduces the high speed response of hydraulic devices The solubility of air in oil depends mainly on the temperature (T) pressure (p) and the density (ρ) of the oil

Of various liquids (water Industrial oil 12 transformer oil and kerosene) kerosene dissolves the greatest amount of air water dissolves the least Of the mineral oils Industrial oil 12 dissolves the least air

If a of
the oil ool
it may

~~The~~ compressibility factor of an oil is characterized by its elasticity of bulk and is determined by the equation

$$\beta_T = \frac{1}{\rho_T}\left(\frac{d\rho}{dp}\right)_T \text{ sq cm per kgf} \qquad (75)$$

where ρ — density kg per cu cm
p — pressure kgf per sq cm
T — temperature °C

Since $\rho = \frac{M}{V}$ where M = mass in kg and V — volume in cu cm then

$$\frac{d\rho}{\rho} = \frac{d\left(\frac{M}{V}\right)}{\frac{M}{V}} = -\frac{dV}{V}$$

and equation (75) can also be written as

$$\beta_T = -\frac{1}{V} \times \frac{dV}{dp} \text{ sq cm per kgf} \qquad (76)$$

The compressibility factor is reduced with an increase in pressure.
Oil density depends mainly upon the pressure and equals

$$\rho_T = \rho_0 (1 + Cp + Dp^2)_T \qquad (77)$$

where ρ_0 = density at atmospheric pressure and the initial temperature T
C and D = certain constants (coefficients) depending upon the temperature

Investigations show that under the service conditions of machine tool hydraulic drives, at $T = 50°$ to 60°C

$C = 6.28 \times 10^{-5}$ sq cm per kgf and $D = 1.14 \times 10^{-7}$ cm^4 per kgf^2

Then

$$\left(\frac{d\varrho}{dp}\right)_T = \varrho_0 (C + 2Dp)$$

and

$$\beta_T = \frac{C + 2Dp}{1 + Cp + Dp^2} \tag{78}$$

Hydraulic drives are most widely used in machines requiring a pressure in the range of $p = 70$ to 200 kgf per sq cm as in machine tools and presses. In this pressure range the oil compressibility factor can be taken, on the average, as

$$\beta_T = 5.8 \times 10^{-5} \text{ sq cm per kgf}$$

The specific change in capacity of a system due to oil compressibility will be (see equation 76)

$$\left(\frac{\Delta V}{\Delta p}\right)_T = -V\beta_T \tag{79}$$

The compressibility factor of synthetic hydraulic fluids, based on water, phosphate esters, etc., is less than that of oil and is equal to

$$\beta_T \approx 3.5 \times 10^{-5} \text{ sq cm per kgf}$$

Resinous deposits in the oil passages of all hydraulic devices (gumming up) are observed when any oil is used. This reduces oil flow and, in the course of time, completely stops flow through small passages (sometimes several microns in size) in components of the control devices in the hydraulic system.

Gum formation depends upon the properties of the oil, the material in which the passage is made, the shape of the passage hole, oil flow rate and its temperature. Other conditions being equal, transformer oil has the greatest tendency to form gum. The gumming-up of a passage is the more intensive with a greater pressure drop, or higher temperature of the oil. Gumming can be eliminated by imparting axial or rocking oscillation to one of the mating parts forming the passage at a frequency up to 30 cps and an amplitude up to 0.02 mm.

CHAPTER 9

HYDRAULIC FLUID POWER LINES

9-1. Piping Design and Installation

Distinction is made between *simple* and *complex* hydraulic fluid lines which may be short or long

In a simple line the diameter d and the flow rate Q of oil are constant values and no oil is delivered along the way to consumers, i e oil discharge is only of the transmission type Complex fluid lines consist of an arrangement of simple lines connected in series in parallel or in a combination of the two

The greater part of the pressure loss in a short line is due to local resistances to flow Machine [illegible] consequently, local resis

Linear pressure losses

$l > 100\ d$, where d is the inside diameter of the piping

The pressure losses due to local resistances can be determined fairly accurately in cases when the distance between adjacent resistances is at least $20\ d$, if it is less, the losses can be determined only approximatoly

No standards or recommendations exist to limit the permissible pressure losses They are difficult to foresee because much depends upon how well tho installation job has been done, upon the amount and types of hydraulic apparatus employed, the number of pipe fittings, their quality, etc

In designing a hydraulic fluid line, the length l of the line, the flow rate Q of the fluid passing through the line, and its density must be given The values to be calculated are the pipe diameter d and the pressure loss Δp Diameter d is determined on the basis of the fluid flow rate Q and flow velocity in the piping, then the pressure loss Δp is calculated Oil flow velocities in the lines of machine tool hydraulic systems are selected in the range from 1 to 7 m per sec, depending upon the length of the line and pipe diameter, conditions under which the hydraulic apparatus is installed, the number of local resistances and the shape of the pipe fittings A flow velocity of $v = 6$ to 7 m per sec is suitable for holes in jointing surfaces, in control valves and in distribution devices In all other cases, a value $v \leqslant 3$ to 4 m per sec is recommended, depending upon the pipe diameter Thus for a pipe diameter $d = 12$ to 50 mm, the recommended range is $v = 3$ to 3 5 m per sec, while for $d > 50$ mm, the velocity should be $v = 3$ 5 to 4 m per sec

The inside diameter of the line is found from the formula

$$d_0 \cong 4.6\sqrt{\frac{Q}{v}} \text{ mm} \tag{80}$$

where Q = flow rate, litres per min
v = oil velocity, m per sec.

As a rule, the hydraulic systems of machine tools have short fluid lines and, therefore, only pressure losses due to local resistances are determined, i.e. as a fraction of the velocity head. Thus

$$\Delta p = K_0 \gamma \frac{v^2}{2g}$$

but, since $v = \frac{Q}{f}$, then $\Delta p = K_0 \gamma \frac{16Q^2}{2g\pi^2 d_0^4}$ or, finally,

$$\Delta p = 4.5 \times 10^4 \gamma K_0 \frac{Q^2}{2gd_0^4} \text{ kgf per sq cm} \tag{81}$$

where γ = weight per unit volume (specific weight) of oil, kgf per cu cm
g = acceleration of gravity, m per sec^2
Q = flow rate, litres per min
d_0 = inside diameter of hydraulic fluid line, mm
K_0 = dimensionless total coefficient of local pressure losses in the pressure and return lines, determined by the formula

$$K_0 = \left[K_1 + \left(\frac{F_2}{F_1}\right)^2 K_2\right] a \tag{82}$$

where a is the correction factor for the degree of turbulence

$$a = \left(\frac{2{,}000}{\text{Re}}\right)^{0.28} \tag{83}$$

where K_1 = total coefficient of local pressure losses in the pressure line
K_2 = total coefficient of local pressure losses in the return line
F_2 = rod end area of the power piston
F_1 = head end area of the power piston
Re = Reynolds number.

It has been shown by various experiments that the values of the coefficients of local pressure losses for oil differ only slightly from those for water which are listed in many handbooks.

In some cases, in designing the hydraulic system of a machine tool with divided flow, it will prove more expedient to use equation (80), but in the generalized form. Thus, introducing the notation

$$4.5 \times 10^4 \frac{\gamma K_0}{2gd_0^4} = R_1$$

we can write

$$\Delta p = R_1 Q^2 \qquad (81\,)$$

where the dimension of Q is litre per min

The coefficients ζ of local resistances are determined experimentally Certain representative values of these coefficients are $\zeta = 0\ 3$ for a pipe bent to a radius $r = (2\ 5 \text{ to } 5)\ d$ for screwed elbows $\zeta = 0\ 72$ $\zeta = 10\ 0$ for a ball valve and $\zeta = 2\ 5$ for a check (nonreturn) valve Adding together *the coefficients of local resistances separately for the pressure and return* lines we obtain the total coefficients of local pressure losses K_1 and K_2 Thu

$$K_1 = \zeta_1 + \zeta_2 + \zeta_3 + \qquad \text{and } K_2 = \zeta_1 + \zeta_2 + \zeta_3 +$$

After substituting these values of K_1 and K_2 in equation (82) we deter mine the total coefficient K_0 of local pressure losses

The flow rate through tho passages in hydraulic apparatus is determined from the formula

$$Q = \mu f \sqrt{\frac{2g}{\gamma}(p_1 - p_2)} \qquad (84)$$

where μ = discharge coefficient for oil $\mu = f\,(\mathrm{Re})$ in tho Reynolds number range $\mathrm{Re} = 40$ to $40\ 000$ $\mu = 0\ 60$ to $0\ 65$ for holes with sharp edges and $\mu = 0\ 8$ to $0\ 9$ for holes with rounded or blunted edges The form of the cross section does not appreciably affect the discharge coefficient (it is almost identical for instance for a narrow slit and for a clearance in the form of an annular slit)

f = cross sectional area of the oil passage

$p_1 - p_2$ = pressure drop in the passage

In calculation of flow through throttles μ is taken from 0 72 to 0 74

The hydraulic systems of machine tools are assembled of seamless steel cold drawn tubing or hot rolled pipe Copper tubing according to USSR Std GOST 617 53 is used in the Soviet Union for interior installation in crowded places Pipes (or tubes) are connected together or to hydraulic components by means of various fittings that comply with government stand ards or the standards of one branch of industry (ENIMS standards for machine tools in the USSR)

Cold drawn tubing is used for lines with an outside diameter $d_o \leqslant 30$ mm hot rolled pipo for $d_o \geqslant 30$ mm The pipe and tubing are made of steel 10 or 20 according to USSR Std GOST 1050 60

Flared type tube fittings (Fig 76) are used in connecting copper and thin walled steel tubing (with a wall thickness $\delta \leqslant 1\ 6$ mm) according to USSR Std GOST 8734 58 and 8732 58 The maximum permissible pressure for copper tubing is $p_{max} = 80$ kgf per sq cm

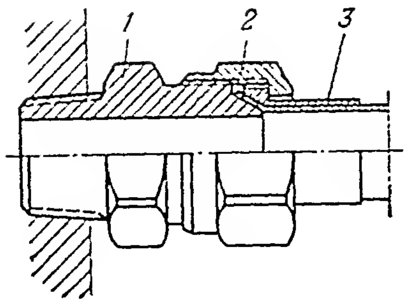

Fig. 76. Flared-type tube fitting:
1—connection; 2—nut; 3—support sleeve

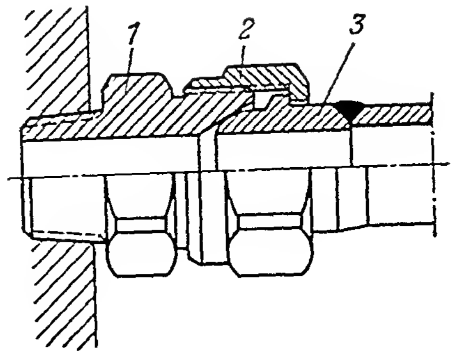

Fig. 77. Welded-type pipe fitting:
1—connection; 2—nut; 3—ball-end sleeve

Welding (Fig. 77) is used to join thick-walled steel pipe ($\delta \geqslant 2$ mm) to the fitting. For pressures of $p \leqslant 200$ kgf per sq cm, pipe of steel 10 with an outside diameter $d_o \leqslant 63$ mm and wall thickness $\delta \leqslant 8$ mm is used; for pressures $p \leqslant 320$ kgf per sq cm, pipe of steel 20 is used with the same d_o and δ dimensions.

Welded-flange fittings (Fig. 78) are used for pipe in the size range $d_o > 63$ mm. If the pipe is of steel 20, the wall thickness will range from 6 to 20 mm, in accordance with the pressure.

Threaded piping is sometimes used for direct connection to components (Fig. 79) in crowded places to facilitate installation.

Steel tubing or pipe is cut off to the required length in a special machine which also makes a chamfer 45° $\times$ 0.5 mm for the weld. Joints are made by d-c welding with a 3-mm electrode, grade УОНИ 13-45. It is advisable to galvanize the thread of fittings to protect them against corrosion.

Tubing is bent in special pipe or tube bending machines. Tubes with an outside diameter $d_o \leqslant 30$ mm and thickness $\delta \leqslant 2$ mm are bent cold without filling them with sand or other supporting material. Tubing of larger diameter is filled with dry sand before bending. Tubing or pipe is tested with hydrostatic pressure before being installed in the machine tool. Sound tubes and pipe are sealed at both ends with plugs to keep them clean inside. Scale and sand that may remain in piping after bending and welding must be carefully cleaned out.

The application of considerable force in installing piping must be avoided as it may lead to the distortion of the thread on the pipe, fittings or apparatus.

On the basis of USSR Std GOST 356-52 (now superseded by GOST 356-59), the ENIMS Institute worked out standard H43-1 stipulating the convention-

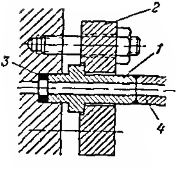

Fig 78 Welded flange pipefitting
1—connection 2—flange
3—packing 4—pipe

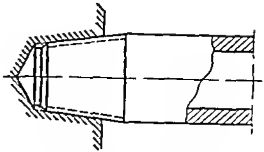

Fig 79 Threaded piping connection

al working and test pressures for pumps hydraulic motors power cylinders and hydraulic apparatus

All of the control and adjusting apparatus is designed to withstand the *conventional pressure* (p_c) on the assumption that it is not subject to surges and other shocks Pumps intended for continuous operation are usually designed on the basis of the *working pressure* (p_w) In testing the strength of fluid power lines pump bodies and other apparatus the *test pressure* (p_t) is used It is assigned in accordance with the conventional pressure and at $2.5 < p_c \leqslant 500$ kgf per sq cm p_t being from 60 to 30 per cent higher than p_c

9-2 Volumetric Changes in the Hydraulic System

The forces that develop in the hydraulic system are applied to the working fluid and are transmitted to the piping The piping is deformed and its capacity is changed The rate at which the deformation takes place is very low Therefore the inertia of the mass being deformed can be neglected and any intermediate position can be regarded as being in equilibrium

The work done by the internal elastic forces is equal to that of the external forces that are converted into potential energy of the deformed system Upon variations in the load the accumulated potential energy is capable under definite conditions of changing the speed and acceleration of the hydraulic motor in the system and of violating its steady motion This promotes the development of vibration in the hydraulic system

The work done in compressing the fluid is

$$A = \frac{\Delta p \, \Delta V}{2} = \frac{(p - p_0)^2 1}{2} \beta_T \text{ kgf-cm} \tag{85}$$

where Δp = pressure increment in the system, kgf per sq cm
V = volume in cu cm of the oil in the system on the delivery side which equals one half the volume of the power cylinder (in a rectilinear motion drive) plus the volume of the delivery line plus one half the pump delivery per revolution (in a rotary drive, the volume V includes one half the hydraulic motor input per revolution instead of one half the cylinder volume)
p and p_0 = final and initial pressures, respectively, in the system, kgf per sq cm
β_T = oil compressibility factor (see Sec. 8-2).

This work is converted into heat, raising the oil temperature.

Oil is heated to an even greater degree by throttling. These changes in oil temperature effect corresponding changes in the volume of the system. Moreover, as a result of volumetric losses from leakage, the amount of oil circulating in the system changes and affects the volume of the system.

The capacity of a hydraulic system to offer resistance to acting forces is called its *rigidity.*

If the volume of a hydraulic system changes due to the action of elastic forces, this change characterizes the *dynamic rigidity* of the system. The smaller the accumulated potential energy of the deformed mass, the larger the dynamic rigidity.

The volumetric losses of the circulating oil characterize the *kinematic rigidity* of the system. The lower this index, i.e. the higher the leakage, the more nonuniform the operation of the hydraulic motor will be.

It is not, however, the volumetric losses that are of prime importance in evaluating the kinematic rigidity of a system, but the variation in these losses in accordance with the operating conditions and the nature of the load.

Volumetric losses due to leakages in the system are determined by the equation

$$q_\sigma = \sigma p_p \text{ cu cm per min} \tag{86}$$

where σ = total coefficient of volumetric losses in the system (pump, control panel and hydraulic motor), cm³/min per 1 kgf/cm²
p_p = pump pressure, kgf per sq cm.

Below are given the values of the coefficient of volumetric losses σ for certain types and sizes of Soviet-made pumps.

Rotary piston pumps

Model	709	712	715
σ, cm⁵/kgf-min	64	133	320

Vane pumps, model Г12, with $p < 65$ kgf/cm²

Q, litres/min =	5	8	12	18	25	35	50	70	100	200
σ, cm⁵/kgf-min =	49.5	46.5	55	72	69	77	108	123	123	233

Gear pumps, model 11 22, with $p \leqslant 25$ kgf/cm²

Q, litres/min =	12	18	25	35	50	70	100	125
σ, cm⁶/kgf min =	144	160	256	300	366	500	560	480

The volumetric losses in the hydraulic apparatus, control panels and power cylinders are substantially smaller (only one per cent or even less) than those in pumps

Elastic deformation of a hydraulic system is made up of the compression of the oil and the distortion of the piping It follows from equation (76) that the reduction in the volume of oil is

$$\Delta V = -V\beta_T \Delta p \text{ cu cm} \tag{87}$$

where V = volume of the piping and passages in the hydraulic system at $p = 0$, cu cm

K = bulk modulus of elasticity of the oil

Δp = pressure increase in the system, kgf per sq cm

$$K = \frac{1}{\beta_T} \cong 1\ 6 \times 10^4 \text{ kgf per sq cm}$$

The piping is deformed to the maximum extent in the radial direction

The relative change in the oil density in the piping is determined from the equation [see also equation (75)]

$$p - p_0 = \Delta p = \frac{\varrho - \varrho_0}{\varrho_0} K = \frac{\Delta \varrho}{\varrho_0} K$$

where p and p_0 are the final and initial pressures, respectively, in the piping.

The change in the radius of the pipe cross section under the action of the pressure, corresponding to this value of pressure drop Δp, is

$$R - R_0 = \Delta R = R_0^2 \frac{p - p_0}{\delta E} = R_0^2 \frac{\Delta p}{\delta E}$$

where R_0 and R = initial and final radii, respectively, of the pipe

δ = pipe wall thickness

E = modulus of elasticity of the pipe material.

The work done in compressing the oil in the piping [see equation (85)] is

$$A_{op} = \frac{\pi R_0^2 l}{\varrho_0} \int \Delta p \, d(\Delta \varrho)$$

where l is the length of the pipe.

After substituting

$$d(\Delta \varrho) = \frac{\varrho_0}{K} d(\Delta p)$$

and integrating, we obtain

$$A_{op} = \frac{\pi R_0^2 l}{2K} (\Delta p)^2 \text{ kgf-cm}$$

The compressibility constant of the oil in the same piping is

$$\Theta=\frac{\pi R_0^2 l}{2K} \text{ cm}^5 \text{ per kgf}$$

and the reduction of the volume of oil in the piping is

$$\Delta V=\Theta\Delta p \text{ cu cm} \tag{88}$$

The work done in expanding the walls of the piping is

$$A_p=2\pi R_0 l\int \Delta p\, d(\Delta R) \text{ kgf-cm}$$

Substituting $\Delta R=\frac{R_0^2}{\delta E}\Delta p$ (see above) and integrating, we have

$$A_p=\frac{\pi R_0^3 l}{\delta E}\Delta p^2=\Theta_1(\Delta p)^2 \text{ kgf-cm}$$

where Θ_1 is the elastic constant of the piping, and is equal to

$$\Theta_1=\frac{\pi R_0^3 l}{\delta E} \text{ cm}^5 \text{ per kgf}$$

Consequently, the increase in the volume of the piping due to the deformation of its walls is

$$\Delta V_p=\Theta_1\Delta p \text{ cu cm} \tag{89}$$

Then, the work done to produce the elastic deformation of the hydraulic system (compression of the oil plus the change in thickness of the pipe walls) is

$$A_0=A_{op}+A_p=(\Theta+\Theta_1)(\Delta p)^2=\Theta_0(\Delta p)^2 \text{ kgf-cm} \tag{90}$$

where $\Theta_0=\Theta+\Theta_1$ is the total compressibility constant of the system. The corresponding change in the volume of the hydraulic system is

$$\Delta V_0=\Delta V+\Delta V_p=(\Theta+\Theta_1)\Delta p=\Theta_0\Delta p \text{ cu cm} \tag{91}$$

The compressibility constant of the oil in the power cylinder, if the drive is for rectilinear motion, and Θ_1, the elastic constant of the material used for the walls of the power cylinder, are determined from the following equations:

$$\Theta=\frac{\pi R_0^2}{2K}=\frac{l_1F_1}{2K};\quad \Theta_1=\frac{\pi R_0^3 l_1}{\delta_1 E}=\frac{R_0 l_1 F_1}{\delta_1 E}$$

where R_0 = radius of the cylinder
l_1 = length of the cylinder
δ_1 = cylinder wall thickness
F_1 = cross-sectional area of the cylinder.

The total potential energy of the hydraulic system is determined by employing equation (90) Thus

$$E_p = (\Theta_0 + cF_1 l_1)(\Delta p)^2 \text{ kgf-cm} \tag{92}$$

where

$$c = \left(\frac{1}{2K} + \frac{R_0}{\delta_1 E}\right) \text{ sq cm per kgf}$$

and

$$\Theta_0 = (\Theta + \Theta_1) \text{ cm}^5 \text{ per kgf}$$

The ratio of the compressibility constants Θ and Θ_1 of the oil and piping is

$$\frac{\Theta}{\Theta_1} = \frac{\pi R_0^2 l \delta E}{2K\pi R_0^3 l} = \frac{\delta E}{2KR_0}$$

Thus, the compressibility constant Θ of the oil is $\frac{\delta E}{2KR_0}$ times as large as the compressibility constant Θ_1 of the piping

For example, if $\delta = 0\,2$ cm, $E = 2 \times 10^6$ kgf per sq cm, $K = 1\,6 \times \times 10^4$ kgf per sq cm and $R_0 = 1\,3$ cm, then

$$\frac{\Theta}{\Theta_1} = \frac{0\,2 \times 2 \times 10^6}{2 \times 1\,6 \times 10^4 \times 1\,3} = \frac{0\,2 \times 10^2}{2\,08} \simeq 10$$

In certain cases (depending upon the piping diameter and wall thickness), the elastic constant Θ_1 of the piping can be disregarded

Equation (90) finds application in calculating the rigidity of a hydraulic system and its natural frequency and in investigations of unsteady motion Equation (91) is used to determine the time lag, or response time, of hydraulic apparatus

If equation (91) is written in the form

$$\frac{\Delta p}{\Delta t} = \frac{1}{\Theta_0}\frac{\Delta V_0}{\Delta t} = \frac{1}{\Theta_0} Q \text{ kgf per cm}^2 \text{ min}$$

where Q is the flow rate of oil, cu cm per min, delivered to the given hydraulic device

It follows that

$$t = \frac{\Theta_0}{Q}(p_2 - p_1) \text{ min} \tag{93}$$

Thus, the larger the compressibility constant Θ_0 of the system, the longer, all other things being equal the response time of the hydraulic apparatus

The velocity of pressure pulse propagation in a hydraulic system depends upon the diameter of the piping and the oil viscosity, but is not affected whatsoever by the pressure Hence at a low viscosity (3 to 5°E), this velocity is $v = 1{,}050$ to $1\,150$ m per sec while at a high viscosity, $v = 740$ m per sec for a pipe diameter $d = 5$ mm and $v = 1\,000$ m per sec for $d = 12$ mm

CHAPTER 10

PUMPS

10-1. Types of Positive-Displacement Pumps

Pumps, in which the conversion of energy occurs in the process of displacing the liquid from the working chambers, are called *positive-displacement pumps*. The working chamber of such pumps is a cavity or pocket capable of changing its volume periodically and connected, alternately, to the suction and discharge cavities of the pump. In these pumps the suction and discharge cavities are hermetically separated from each other.

The part of the pump that directly participates in the process of displacing the liquid from the working chambers is called the *impelling element* (piston, plunger, vane, etc.).

Positive-displacement pumps may differ greatly in construction and are classified, in accordance with the nature of the liquid-displacing process they employ, into *reciprocating* and *rotary* pumps. In the former, the liquid is displaced from stationary working chambers, i.e. only due to the reciprocating motion of the impelling elements (pistons or plungers), as in the Soviet-made eccentric-driven pumps, model Г17. In rotary pumps, liquid displacement is a result of rotary or rotary-reciprocating motion of the impelling elements.

Rotary pumps can be divided into the *purely rotary* type, in which the impelling elements only rotate, as in screw and gear pumps, and *rotary piston pumps*, in which the impelling elements (plungers) reciprocate, in addition to rotation, in respect to the rotating cylinder body.

In accordance with the form of the impelling elements and the method resorted to to close off the volume of liquid being displaced, pumps operating with rotary-reciprocating motion are classified as *vane pumps*, for example model Г12, *radial rotary piston pumps* as model Г13, and *axial rotary piston pumps* as model МГ15.

In positive-displacement pumps, the working chamber is formed in the suction cavity as a result of simultaneous motion of the rotor and separating element, or of the rotor and stator. At a definite moment in time, the volume of liquid that has entered the working chamber is closed off and isolated from the suction and discharge cavities of the pump. At the next moment, this volume is squeezed or forced into the discharge cavity. As the liquid is carried from one cavity to the other, there is always a tendency to compress the volume of liquid locked in the working chamber. This

reduces the pump efficiency, heats the liquid and subjects the parts of the pump to an additional load

Valve distribution is rarely used in rotary pumps. In some cases, this function is fulfilled by passages in the central pintle on which the cylinder body rotates.

In rotary piston pumps with axial arrangement of the cylinders in the rotating body, a special disk with holes in an end face is used for distribution. It is called a valve-plate facer. This facer is stationary in certain models, it rotates in synchrony with the cylinder body, or barrel, in others.

All rotary pumps are invertible, i.e. they can operate both as pumps which consume energy, and as hydraulic motors which transmit energy.

10-2. Pump Characteristics and Size Range

In positive-displacement pumps, the delivery should not vary with the load, i.e. the theoretical characteristic $Q_0 = f(p)$ of any positive-displacement pump is a horizontal straight line in a rectangular co-ordinate system (line cH_0 in Fig. 80). However, due to inevitable volumetric losses in the pump, the actual characteristic Q will be a straight line inclined at an angle in respect to the theoretical characteristic.

The higher the volumetric losses in the pump, the larger this angle will be. Operating point K on the actual characteristic $Q = f(p)$ for given operating conditions is at the intersection of the hydraulic resistance characteristic a of the pipeline and the actual pump characteristic. It follows that the required pump pressure p_p, set up by adjusting the spring of the relief valve, is $p_p = p_1 + \Delta p$, where p_1 is the pressure needed to overcome the load of the hydraulic motor and Δp is the pressure loss in the hydraulic system of the machine tool, determined by the piping characteristic [see equation (81)]. The distance q_p on the axis of ordinates is used to determine the control factor ψ of the pump and therefore the minimum speed of the power cylinder piston or shaft of a rotary hydraulic motor. Thus

$$\psi_{min} = \frac{q_p}{K_p n}$$

where K_p = pump constant, cu cm per revolution
n = speed, rpm, of the pump rotor

The length q (in cu cm per min) on the axis of ordinates shows the volumetric losses of the pump at maximum pressure. The volumetric loss coefficient of the pump is $\sigma_p = \frac{q}{p_{max}}$ cm^5/kgf min (i.e. in cm^3/min per 1 kgf/cm^2).

Flow control sharply changes the actual characteristic of a pump. Operat-

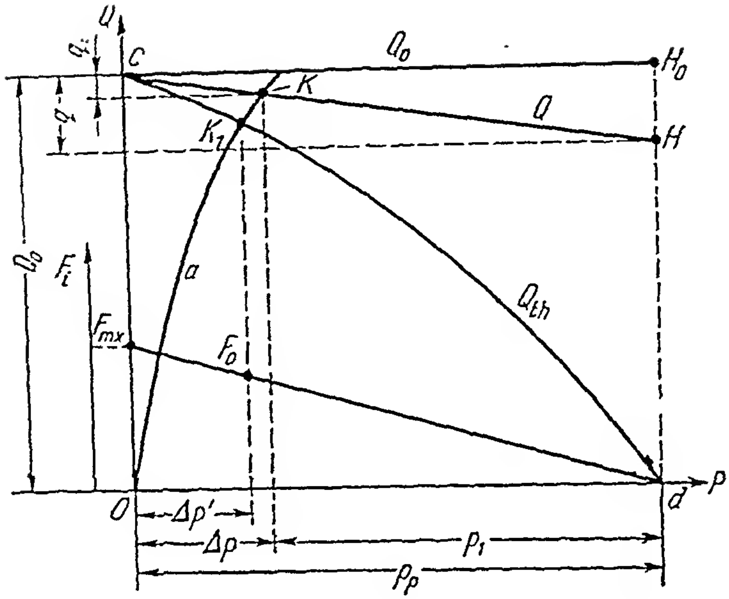

Fig. 80. Characteristic of a positive-displacement pump

ing point K_1, in this case, is the intersection of the hydraulic resistance characteristic *a* of the piping and the flow control valve characteristic Q_{th} (Fig. 80). The area *cHdc* between the characteristics is the surplus delivery of the pump which must be drained back to the tank through the relief valve. This valve maintains a constant pressure p_{max} in the system. Consequently, any type of flow control requires that the volumetric loss characteristic *cH* of the pump be located above the *flow* control characteristic Q_{th} over the whole range of pressure regulation $Od = p_p$.

If the change in the area F_1 of the orifice in the flow control valve is plotted on the same graph, but at another scale, the point corresponding to cross-sectional area F_0 marks the beginning of flow control of the pump delivery. In the interval $F_{max} - F_0$ of orifice area variation, the flow control valve cannot change the pump delivery since the hydraulic resistance $\Delta p'$ in the piping is higher than that of the flow control valve. Therefore, in using flow control, the resistance of the piping must be reduced in order to expand the range of pump delivery regulation. This is done by increasing the diameter of the piping, reducing the number of sources of local losses, etc. The characteristic of the hydraulic resistance in the piping does not depend upon the type or construction of the pump. It depends only upon the dimensions of the piping, the number and types of fittings and control apparatus.

The pressure head in the suction line during pump operation is

$$h = h_a - \frac{\varrho_{oil}}{\varrho} H_s - \zeta \frac{v^2}{2g} - \sum h_l - h_b \text{ m}$$

where h_a = atmospheric pressure, metres of mercury or water column

ϱ_{oil} and ϱ = respective densities of oil and water (or mercury) expressed in the same units

H_s = difference in the levels of the free surface of tho oil in the tank and in the suction chamber of the pump, m

h = pressure head in the suction line, m

v = oil velocity in the suction line, m per sec

g = acceleration of gravity, m per sec²

h_l = pressure loss in the suction line, m

h_b = pressure loss in the pump body, m.

Therefore, the permissible suction lift is

$$H_s = \frac{\varrho}{\varrho_{oil}} \left(h_a - h - \zeta \frac{v^2}{2g} - \sum h_l - h_b \right) \text{ m} \tag{94}$$

At the limiting, or so-called critical, suction height cavitation occurs; the oil begins to generate vapour intensively, the stream of oil in the suction pipe breaks away from the walls, and the cross section of the stream is narrowed

Cavitation sets in at the moment when the pressure in the suction pipe is reduced to the oil vapour tension at the given temperature Accordingly, the critical suction height is determined by the condition $H_{s\text{-}max} = H_{t\text{-}oil}$, where $H_{t\text{-}oil}$ is the pressure of the oil vapour at the given temperature t. The vapour tension is the pressure of a liquid at which, at a given temperature, it begins to boil and free evaporation takes place The vapours of the working fluid, mineral oil in this case, fill all the enclosed space in the suction chamber to the saturation point The vapour tension of oil is very low and this promotes severe infiltration of air into the suction chamber of the pump, especially when the oil is hot Hence, the suction line must be absolutely airtight This is sometimes achieved by immersing the pump in the oil tank

Mineral oil always contains dissolved air in an amount ranging from 7 5 to 13 5 per cent by volume This affects the suction capacity of the pump and its characteristic A certain reduction in liquid delivery is observed in continuous operation of a pump This is associated with foaming of the oil and the formation of an air oil emulsion As a result, the suction chamber is only partly filled with oil Attempts to remove the dissolved air from the oil by the prolonged action of a vacuum have not been successful; the oil is intensively emulsified and its elasticity is changed

The dissolved air has been observed to display greater activity in rotary piston pumps than, for example, in gear pumps. This is probably due to the effect of the dead space in the working chambers of piston pumps.

The formation of a gaseous phase may have two causes: the mechanical entrainment of atmospheric air by the oil during the suction period, and the evolution of the air dissolved in the oil.

There should be no mechanical entrainment of the air if the suction line is absolutely airtight and the oil velocity is within normal limits.

The evolution of dissolved air from the oil in the suction period depends upon the absorption factor ε and is of a regular nature, its magnitude being constant at a given oil temperature. If the factor ε is known, we can determine the amount of air evolved from the oil when the pressure is reduced in the suction chamber. Mineral oils with kinematic viscosities $\nu = 2$, 3.5 and 10 sq cm per sec, have absorption factors $\varepsilon = 0.07$, 0.072 and 0.08, respectively.

In addition to useful work, the power consumed by a pump is expended in overcoming: (a) mechanical losses due to friction in the pump mechanism, (b) pressure losses as the oil flows through the pump channels and the delivery and drain lines, and (c) volumetric losses of the pump. Thus, the power input of the pump is

$$N_p = N + N_V + N_f$$

where N = power used in doing useful work

$$N = \frac{pQ}{A}$$

p = pressure in the hydraulic motor (or cylinder)
Q = oil flow rate through the hydraulic motor
A = conversion factor
N_V = power expended in compensating for volumetric losses in the pump

$$N_V = \frac{p_p q}{A}$$

$p_p = p + \Delta p$
Δp = pressure losses in the delivery and drain lines of the hydraulic system; see equation (81)
q = volumetric losses, $q = \sigma_1 p_p$; see Fig. 80.

The power N_f, expended in overcoming friction and hydraulic losses in the pump, is determined by the mechanical efficiency of the pump. Thus

$$N_f = \eta_m N$$

in which

$$\eta_m = \frac{N_l}{N_p}$$

where $N_i = \frac{Q_0 p_p}{A\eta_h}$ — indicated power of the pump

Q_0 = geometrical delivery of the pump (see Fig 80)

N_p = power input of the pump which is determined with a wattmeter, taking into account the efficiency of the electric motor The power input can be more accurately determined from the torque measured by an electric dynamometer with a floating frame

η_h = hydraulic efficiency

The power N_f can also be determined as the difference $N_p - N - N_i$, but this will require that the input power N_p of the pump be known

The hydraulic efficiency η_h characterizes the pressure loss in a pump due to oil flow through the internal channels in the pump body Thus

$$\eta_h = 1 - \frac{p_1}{p_0}$$

where p_0 = pressure of the oil as it enters the pump body channels

p_1 = pressure of the oil leaving the channels

The pressure difference $p_0 - p_1 = \Delta p_1$ is determined on a special test stand and is then used in the equation

$$\Delta p_1 = \gamma K_1 Q^2 \text{ kgf per sq cm} \tag{95}$$

where K_1 is the constant depending upon the shape of the channels in the pump body

This constant is similar to the total coefficient in equation (82) Hence, the overall efficiency of the pump equals

$$\eta_0 = \eta_v \eta_m \eta_h$$

The hydraulic efficiency is usually quite high η_h = 0 985 to 0 970 and need therefore be determined only for new models of pumps

Investigations have shown that the power required to compensate for volumetric losses increases with the oil temperature and therefore with the time that the pump is in continuous operation On the other hand the power required to overcome the friction forces is substantially reduced with an increase in oil temperature

The output or delivery is the principal characteristic of a hydraulic pump as are the rate of flow through control apparatus and the displacement of a hydraulic cylinder In the Soviet Union the rates of flow have been standardized by ENIMS beginning with the value 18 litres per min the values constitute a geometrical series with the ratio φ = 1 41

Standard series of pump deliveries in litres per min

0 5 1 2 3 5 8 12 18 25 35 50 70 100 140
200 280 400 560 800 1 120 1 600 2 250 3 200
4 500 6 300 9 000 and 12 000

The minimum delivery of pumps used in the hydraulic drives of machine tools has been taken as $Q_{min} = 3$ litres per min, and the maximum delivery Q_{max} as 400 litres per min.

Proper pump selection, as regards the model, delivery and pressure developed by the pump, are of prime importance in designing hydraulically operated machine tools. Performance of the machine tool may depend upon how correctly this choice has been made.

The machine tool industry of the USSR manufactures a series of extensively used models of constant- and variable-displacement pumps for various deliveries and pressures. These include:

1. Variable-displacement radial piston pumps, model Г13: (a) with a manually controlled displacement and having a delivery range $Q = 50$ to 200 litres per min at a pressure of $p = 200$ kgf per sq cm, model ВГ13-21А: (b) with hydraulic controls and having a delivery range $Q = 30$ to 200 litres per min at a pressure of $p = 75$ kgf per sq cm, model ВГ13-16А*; (c) with electrohydraulic controls and having a delivery range $Q = 15$ to 100 litres per min at a pressure of $p = 100$ kgf per sq cm, model Г13-15А; (d) with a delivery range $Q = 30$ to 200 litres per min at a pressure of $p = 75$ kgf per sq cm, model Г13-16А, and (e) with hydraulic servosystem controls and having a delivery range $Q = 15$ to 50 litres per min at a pressure of $p = 200$ kgf per sq cm, model БГ13-24А.

Rotary piston pumps with reversible delivery are intended for use in the hydraulic systems of heavy machine tools having a high pull capacity and operating at high speeds (slotters, broaching machines, planers, etc.).

2. Constant-displacement nonreversible-delivery vane pumps with a delivery in the range $Q = 3$ to 200 litres per min at a pressure up to $p = 65$ kgf per sq cm, of both the single and double types. Double vane pumps are used for various combinations of deliveries and for various pressures. Pumps with a high delivery are designed for pressures up to 25 kgf per sq cm; those for low delivery—for pressures up to 65 kgf per sq cm.

Vane pumps are widely employed in hydraulic machine tools with flow control in which the operative units travel at low speed and with heavy cutting pressures. Double pumps, also called two-volume, low- and high-pressure pumps, are applied in circuits of machine tools operating with rapid traverse motions and slow working feeds. Here the low-pressure, high-delivery pump is used for rapid traverse.

3. Gear pumps with a delivery in the range $Q = 12$ to 125 litres per min at a pressure of $p = 13$ kgf per sq cm, model Г11-1, and with a delivery in the range $Q = 12$ to 140 litres per min at a pressure of $p = 25$ kgf per sq cm, model Г11-2. Gear pumps find application in the hydraulic systems of

* See the hydraulic circuits of machine tools in Figs. 201, 207, 209 and 213 (Vol. 1).

machine tools operating at medium and low pressures and at high speeds (grinders, honing machines, etc.)

4. Combination pumps: axial piston pumps combined with vane pumps and having the following deliveries: axial pumps—$Q = 3$ to 8 litres per min at a pressure of $p = 100$ kgf per sq cm, and vane pumps—$Q = 25$ to 100 litres per min at a pressure of $p = 25$ kgf per sq cm in any combination of deliveries. Such combination pumps (model Г14) are used in hydraulically operated machine tools operating on an automatic cycle,* and also for various auxiliary devices, for example, to clamp the blank, to clamp the tailstock spindle, etc.

10-3. Pump Operation. Automating the Hydraulic Power Unit

Both single and dual—series or parallel—pump operation methods are used in the hydraulic systems of machine tools. Series pump operation is used to increase the pressure in the system, for instance, during the working travel of the hydraulic motor. Parallel pump operation, in which the delivery is correspondingly increased, is used to obtain rapid traverse motions, such as rapid approach and withdrawal of an operative unit, release of the finished workpiece and clamping of the next blank in handling devices, etc. Separate and simultaneous operation systems of the pumps are widely used in these circuits. The pump is always switched over from one mode of operation to another automatically. The pulse for actuating the automatic controls is usually produced by an increase in pressure in the system; in-travel controls are also frequently used.

A circuit for separate pump operation can be made up of separate standard hydraulic apparatus if the overall dimensions of the machine tool are not specified or the required delivery of the pumps exceeds ~150 litres per min. Otherwise a sequencing control panel (model Г53 in the USSR), containing all the automatic controls, can be used.

The diagram of a circuit made up of standard apparatus for *separate operation of pumps*, is shown in Fig. 81. For rapid forward travel of the piston, the discharges Q_1 and Q_2 from the two pumps are admitted in parallel to the power cylinder along lines *1 2 3 4* and *5 4*, so that the piston moves rapidly to the right. The pressure of the pump at this moment depends upon the friction losses and the backpressure (if provided for in the circuit) and is

$$p_S = \frac{1}{F_1}\left(\sum S + p_2F_2\right) \text{ kgf per sq cm} \tag{96}$$

* See Fig. 183 (Vol. 1) which shows a variable displacement piston pump P_{vd} and two vane pumps P_1 and P_2.

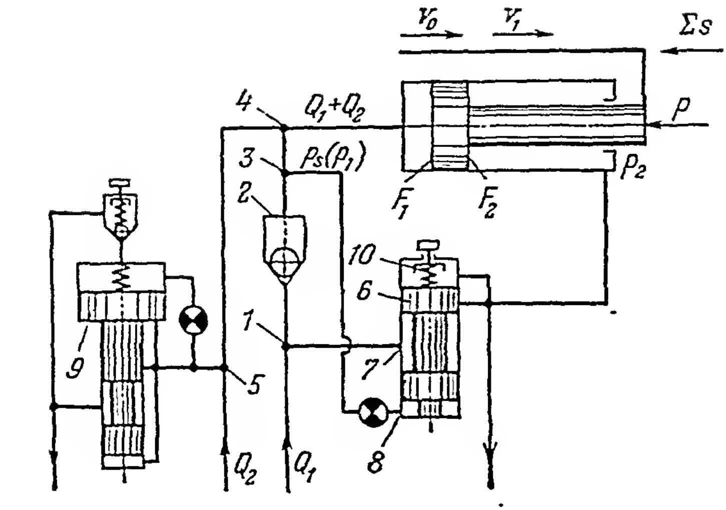

Fig. 81. Circuit for the separate operation of two pumps:
1—line from low-pressure pump discharge; *2*—check valve, model Г51; *6*—backpressure valve, model Г54; *9*—relief valve, model Г52, of high-pressure pump

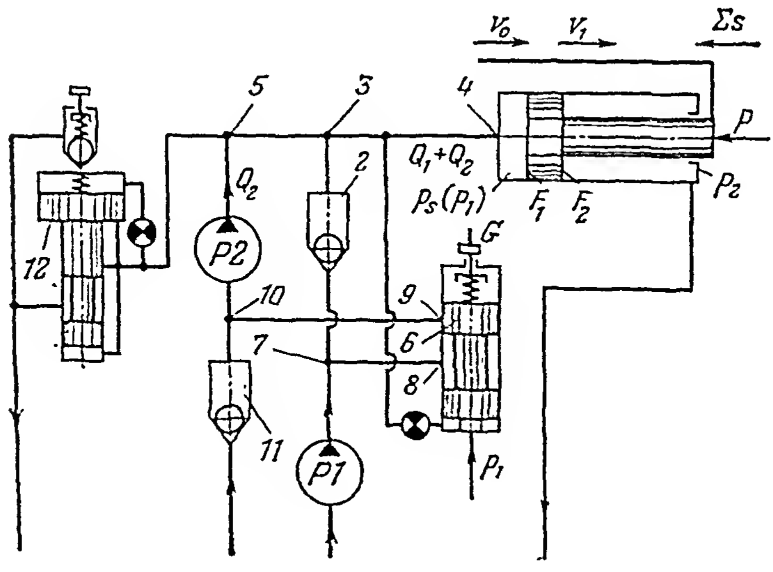

Fig. 82 Circuit for the simultaneous operation of two pumps

where $\sum S$ = total friction force (see Sec 11 3), kgf

p_2 = backpressure in the power cylinder, kgf per sq cm

F_1 and F_2 = head and rod end piston areas, sq cm

The pressure p_S is set by adjusting spring *10* of the low pressure relief valve The force exerted by the spring is found from the equation

$$p_S=\frac{4G_1}{\lambda\pi d_1^2}\text{ kgf per sq cm} \tag{97}$$

where G_1 = force exerted by the spring, kgf

λ = 1 1 to 1 15 = assurance factor

d_1 = piston diameter of relief valve Г54

Combining the last two equations and solving for the required force of the spring we obtain

$$G_1=\frac{\lambda\pi d_1^2 p_S}{4}=\frac{\lambda\pi d_1^2}{4}\frac{1}{F_1}\left(\sum S+p_2F_2\right)\text{ kgf} \tag{98}$$

When the load is applied, the pump pressure increases It also increases in line *3 8* so that check valve *2* closes off discharge Q_1 from the pump Then, under the action of pressure p_1 ($p_1 > p_S$), piston *6* of the low-pressure relief valve is shifted upward, and line *1 7* is connected to the tank

After the load is applied, the pressure in the system is

$$p_1=\frac{1}{F_1}\left(P+\sum S+p_2F_2\right)\text{ kgf per sq cm} \tag{99}$$

where P = load, kgf

$\sum S$ = total friction force when the piston travels under load, kgf

The spring of relief valve *9* of the high-pressure pump is adjusted to a pressure $p_p = p_1 + \Delta p$, where Δp is the pressure loss in the discharge and drain lines [see equation (81)]

During piston travel under load discharge Q_1 of the pump drains back to the tank at almost zero pressure so that the pump consumes only a slight amount of power

Upon rapid traverse of the piston the power required by the hydraulic power unit is

$$N_S=\frac{p_S}{612}(Q_1+Q_2)\text{ kW} \tag{100}$$

where Q_1 and Q_2 are the respective deliveries, litres per min, of the pumps at p_S kgf/cm²

The power required by the same power unit when the piston travels under load, is

$$N_w=\frac{1}{612}(p_SQ_1+p_1Q_2)\text{ kW} \tag{101}$$

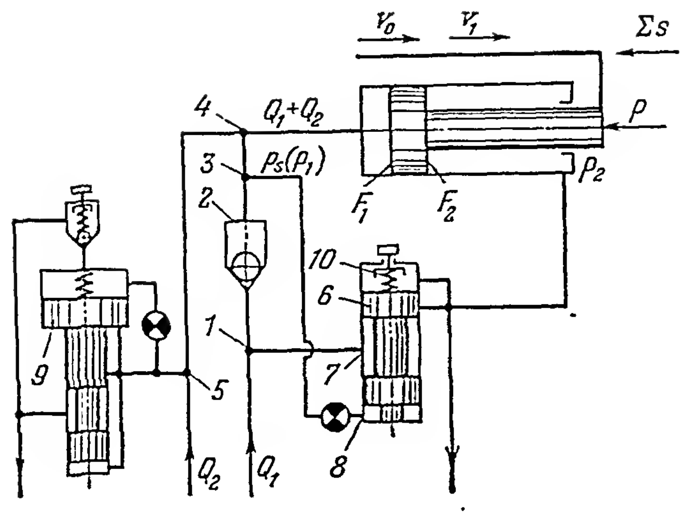

Fig. 81. Circuit for the separate operation of two pumps:
1—line from low-pressure pump discharge; 2—check valve, model Γ51; 6—backpressure valve, model Γ54; 9—relief valve, model Γ52, of high-pressure pump

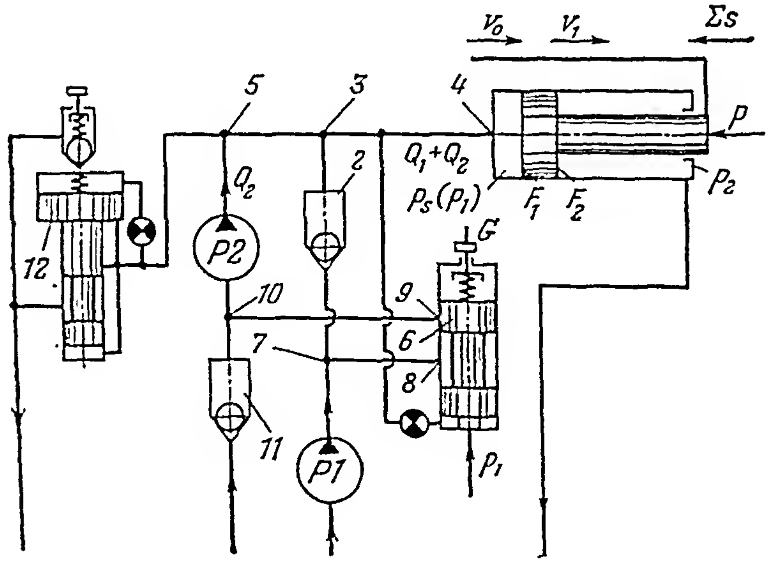

Fig. 82 Circuit for the simultaneous operation of two pumps

where $\sum S$ = total friction force (see Sec 11-3), kgf

p_2 = backpressure in the power cylinder, kgf per sq cm

F_1 and F_2 = head and rod end piston areas, sq cm

The pressure p_S is set by adjusting spring *10* of the low-pressure relief valve The force exerted by the spring is found from the equation

$$p_S = \frac{4G_1}{\lambda \pi d_1^2} \text{ kgf per sq cm} \tag{97}$$

where G_1 = force exerted by the spring, kgf

λ = 1 1 to 1 15 = assurance factor

d_1 = piston diameter of relief valve Г54

Combining the last two equations and solving for the required force of the spring we obtain

$$G_1 = \frac{\lambda \pi d_1^2 p_S}{4} = \frac{\lambda \pi d_1^2}{4} \frac{1}{F_1}\left(\sum S + p_2F_2\right) \text{ kgf} \tag{98}$$

When the load is applied, the pump pressure increases It also increases in line *3 8* so that check valve *2* closes off discharge Q_1 from the pump Then, under the action of pressure p_1 ($p_1 > p_S$), piston *6* of the low pressure relief valve is shifted upward, and line *1 7* is connected to the tank

After the load is applied, the pressure in the system is

$$p_1 = \frac{1}{F_1}\left(P + \sum S + p_2F_2\right) \text{ kgf per sq cm} \tag{99}$$

where P = load, kgf

$\sum S$ = total friction force when the piston travels under load, kgf.

The spring of relief valve *9* of the high pressure pump is adjusted to a pressure $p_p = p_1 + \Delta p$, where Δp is the pressure loss in the discharge and drain lines [see equation (81)]

During piston travel under load, discharge Q_1 of the pump drains back to the tank at almost zero pressure so that the pump consumes only a slight amount of power

Upon rapid traverse of the piston the power required by the hydraulic power unit is

$$N_S = \frac{p_S}{612}(Q_1 + Q_2) \text{ kW} \tag{100}$$

where Q_1 and Q_2 are the respective deliveries, litres per min, of the pumps at p_S kgf/cm²

The power required by the same power unit when the piston travels under load, is

$$N_w = \frac{1}{612}(p_SQ_1 + p_1Q_2) \text{ kW} \tag{101}$$

where p'_S is the pressure of the pump with delivery Q_1 when the piston travels under load. As a rule

$$p'_S = 0.5 \text{ to } 0.8 \text{ kgf per sq cm}$$

The diagram of a circuit for *simultaneous operation of pumps*, assembled of standard hydraulic apparatus, is shown in Fig. 82. Special control panels for simultaneous pump operation are not manufactured in the Soviet Union.

The pressure setting p_G for spring G is determined from equation (97) and, as long as it exceeds the rapid traverse pressure p_S, both pumps, *P1* and *P2*, deliver oil to the power cylinder. Pump *P1* delivers oil through line *2-3-4* and pump *P2*, through line *5-3-4*. Thus, at point *3*, the discharges of the two pumps are added together.

As the pressure in the system increases, force P_1, acting on the end of the piston of valve *6*, becomes larger than the effort p_G of the spring in the same valve. Consequently, the valve piston is shifted upward, connecting the discharge line *7-8-9-10* of pump *P1* with the suction line of pump *P2* so that the pumps begin to operate in series.

Item *11* in Fig. 82 is a check valve, model Г51, while item *12* is a relief valve, model Г52.

In the series operation of pumps, it is necessary to comply with the condition that the delivery of pump *P1* must be larger than that of pump *P2* by an amount at least as large as the volumetric losses of pump *P2*. Thus

$$Q_1 > (Q_2 + \sigma_2 \Delta p) \text{ litres per min} \tag{102}$$

where Δp is the increase in pressure in the system, i. e.

$$\Delta p = (p_1 - p_S) \text{ kgf per sq cm}$$

In principle, any increase in pressure can be obtained by connecting several pumps in series. This measure, however, lowers the efficiency of the power unit. The greater the number of pumps operating in series, the lower the overall efficiency of the power unit. This efficiency should never be less than 0.55 or 0.50.

The power requirement for the working stroke is

$$N_w = \frac{1}{612} [p_S (Q_1 - Q_2) + p_1 Q_2] \text{ kW} \tag{103}$$

In order to reduce the required pump discharge and, consequently, the power requirement, some circuits make use of the oil exhausted from the rod end of the power cylinder. Three-position four-way directional valves, model 2Г-73-1 or 2БГ-74-1, with open centre and closed exhaust (Fig. 83) are used. If the piston rod diameter is d, the rapid forward traverse of the piston is

$$v_0 = \frac{4 (Q_1 + Q_2)}{\pi d^2} \tag{104}$$

It is not always necessary however to use oil exhausted from the cylinder to increase the speed of travel The application of this method depends upon the ratio of the rapid traverse speed to the rate of working travel $\lambda = \frac{t_0}{t_1}$ and the ratio of the piston area to the cross sectional area of the rod $\frac{D^2}{d^2}$

It follows from the equation $\frac{Q_1+Q_2}{kd^2} = \frac{Q_1}{D^2}$ that the delivery of the low pressure pump should be

$$Q_2 = Q_1\left(\lambda \frac{d^2}{D^2} - 1\right) \qquad (105)$$

It is evident from the last equation that at $\lambda < \frac{D^2}{d^2}$ the delivery of the low pressure pump is not required

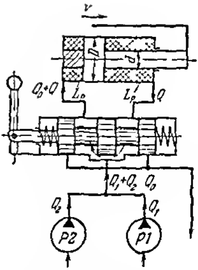

Fig 83 Circuit using exhausted oil for increasing power piston speed

P1 and *P2*—pumps *1*—three-position four way directional valve model 2Г74 1

In some cases the operating cycle of the machine tool requires the provision of intervals in pump operation (during loading and clamping operations blank and work handling setting up motions etc) These intervals are usually effected by one of two methods (1) shutting off pump delivery in which case the whole discharge of the pump is drained to the tank through a relief valve and the power consumed by the pump is converted into heat and (2) by applying a special device which unloads the discharge at practically zero pressure to the tank

Shutting off pump discharge is economically feasible only for short intervals or at relatively low pressures With prolonged intervals and/or high pressures such a circuit is subject to excessive heating of the oil and a substantial drop in efficiency

If directional control valve model 6Г 1 (Fig 84*a*) is used for this purpose the ends of the power cylinder will be shut off in the central position of the valve and the whole discharge of the pump will be unloaded to the tank through the longitudinal channel in the valve piston at almost zero pressure

If the whole system is to be unloaded in this way use can be made of directional control valve model Г73 1 or Г74 1 (Fig 84*b*) with an open centre and all ports open to exhaust

In connection with the fact that oil as any other liquid is very responsive to volumetric losses due to low compressibility it is necessary to develop

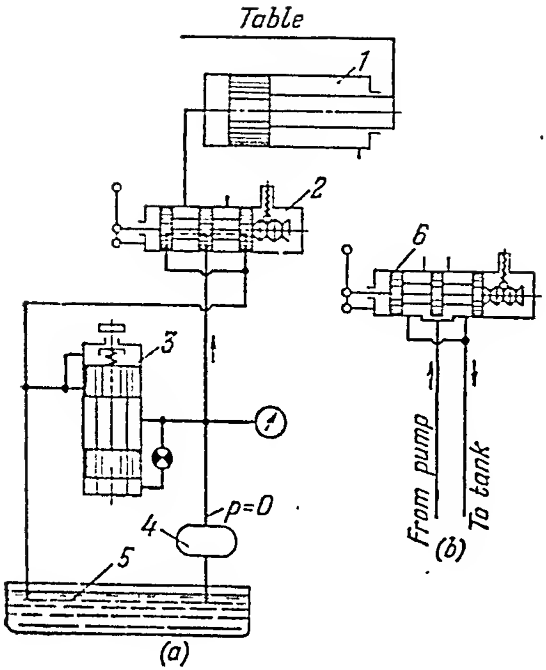

Fig. 84. Circuit for unloading pump discharge through a directional control valve:

1—power cylinder; *2*—directional control valve, model 6Г74-1; *3*—backpressure valve, model Г54; *4*—pump; *5*—tank; *6*—directional control valve, model 1Г74-1

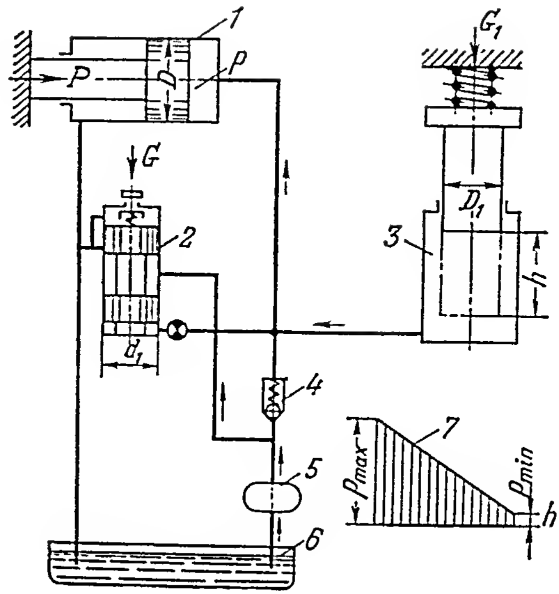

Fig. 85. Circuit for automatic unloading of pump discharge by means of an accumulator:

1—power cylinder; *2*—backpressure valve, model Г54; *3*—spring-type accumulator; *4*—check valve, model Г51; *5*—pump; *6*—tank; *7*—accumulator spring characteristic

a pressure for compensating losses of volume in a system used for clamping operations. These losses are very small compared with the discharge of the pump, so that the whole delivery will be drained to the tank at full pressure, the consumed power being converted into heat. For this reason, a more expedient circuit is one in which the pump is switched in *periodically* as the pressure drops in the system. The circuit shown in Fig. 85 incorporates a spring-type accumulator *3* which responds to changes of pressure in the system.

The required clamping pressure P is set up by adjusting the spring of relief and unloading valve *2* (model Г54). The force exerted by the spring should be

$$G = P\frac{d_1^2}{D^2}$$

where d_1 = diameter of the piston of valve *2*
D = diameter of the clamping piston

In a similar way, the force exerted by the accumulator spring should be

$$G_1 = P\frac{D_1^2}{D^2}$$

where D_1 is the diameter of the accumulator plunger.

When the accumulator is charged to maximum pressure p_{max}, unloading valve *2* is operated, bypassing the discharge of the pump to the tank at zero pressure. This continues until the pressure drops to the minimum value p_{min}. The larger the volume of the accumulator, the less frequently the pump will be cut into the circuit. The minimum volume of the accumulator, the volume of a single charge, is

$$V_1 = \frac{\pi D_1^2}{4} h = \frac{p_{max} - p_{min}}{c}\left(\frac{\pi}{4} D_1^2\right)^2 \text{ cu cm}$$

where p_{max} = maximum pressure in the system which usually does not exceed 12 kgf per sq cm
p_{min} = minimum pressure in the system, kgf per sq cm
c = rigidity of the accumulator spring, kgf per cm.

In air- or gas operated accumulators (Fig. 86), the air (or gas) is usually separated from the oil by a diaphragm, floating piston *5* or a rubber bladder. Sometimes accumulators with direct contact of the two media are employed, it is advisable to use nitrogen (a neutral gas) instead of air in such cases.

The accumulator has a valve box *3* with two check valves, *1* and *2*. Oil from the pump is admitted through valve *2* into bottle *4* of the accumulator, and is discharged into the system through valve *1*.

Air- or gas operated accumulators of this design have a great amount of dead space since full discharge of the oil is impossible as the gas may escape into the working cylinder. This condition is prevented by limiting the minimum pressure, selecting the spring of valve *1* properly for this purpose. Then if the pressure drops below the preset value, the accumulator is automatically cut out of the system.

The minimum permissible pressure in the system is

$$p_{min} = \frac{p_{max}}{1.1 \text{ to } 1.2} \tag{106}$$

Thus, it can be seen that only about 20 per cent of the volume of the accumulator is utilized. The maximum pressure of the accumulator is limited by the spring of valve *6* (model Г54).

While the accumulator is being charged, the pressure line to the power piston is cut off. The pump is cut in and it delivers oil through check valve *7* (model Г51) into valve box *3* and accumulator bottle *4*. The pressure increases, and floating piston *5* moves upward, compressing the air in the upper

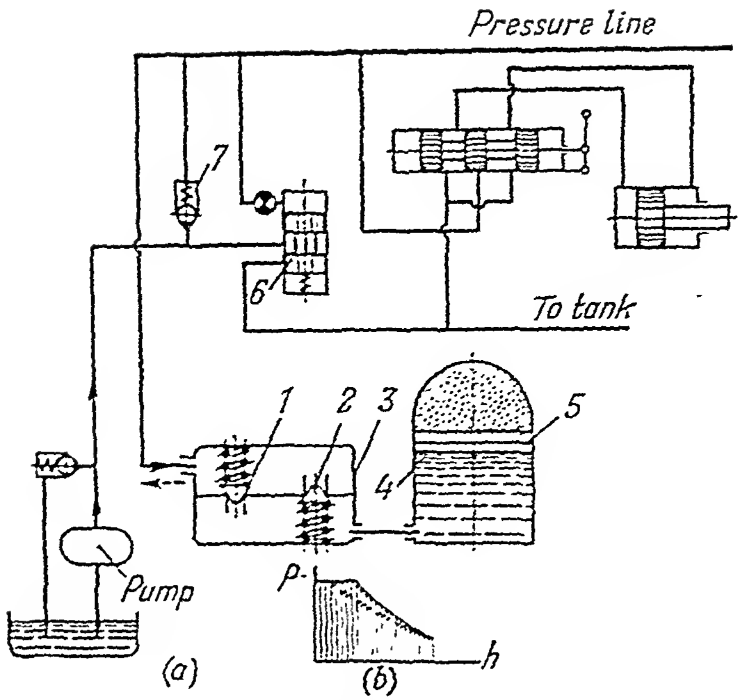

Fig. 86. Circuit for unloading pump discharge by means of an air-operated accumulator (*a*) and the accumulator characteristic (*b*):
5—valve box; *6*—backpressure valve, model Г54, for cutting out the pump

end of the bottle. When the preset pressure is reached, valve *6* is actuated, the pressure line is closed off by check valve *7*, and the pump is switched over to the tank to which oil is delivered at almost zero pressure.

If the pressure end of the accumulator is connected to the pressure line leading to the power cylinder, the accumulator will maintain the given pressure, compensating for volumetric losses in the system in the range $p_{max}-p_{min}$. When the pressure falls below p_{min}, valve *6* returns to its initial position, the pump is automatically cut in and the pressure is raised to p_{max}.

A circuit of this type can be used, not only for various clamping facilities, but in cases when it is necessary to increase the fast-response of a hydraulic motor (power cylinder or rotary motor), for example, in engaging a clutch, etc.

The minimum volume of air required in the accumulator bottle is

$$Q_A = \frac{q}{\sqrt[k]{1-\varsigma}-1} \qquad (107)$$

where q = required volume of liquid in the accumulator

$\varphi = \frac{p_{max} - p_{min}}{p_{min}}$ = degree of pressure nonuniformity in the system, usually equal to 0.1-0.2

z = polytropic curve factor of air compression and expansion determined from the equation $pQ_A^z = \text{const}$, i.e.

$$z = \frac{\log \frac{p_{max}}{p_{min}}}{\log \frac{Q_{A\,max}}{Q_{A\,min}}} \tag{108}$$

In the equations for φ and z, the letter p denotes the pressure in the accumulator and Q_A denotes the volume of air in the bottle at the corresponding pressure. The polytropic curve factor z is within the limits $1 < z < 1.4$.

The required volume of the bottle of an air operated accumulator is

$$Q = q + Q_A = q\left(1 + \frac{1}{\sqrt[z]{1+\varphi} - 1}\right) \tag{109}$$

For tentative calculations, it can be assumed that

$$Q_A \cong (8 \text{ to } 10)\,q$$

then

$$Q \cong (9 \text{ to } 11)\,q \tag{110}$$

CHAPTER 11
HYDRAULIC CYLINDERS AND ROTARY MOTORS

11-1. Types of Hydraulic Cylinders

The hydraulic cylinder is a positive-displacement reciprocating hydraulic motor which converts the energy of a liquid into the kinetic energy of the moving piston or cylinder.

Hydraulic cylinders have found widespread application in many types of machinery, including machine tools, due to their simple construction, high dependability in operation, comparatively low cost and the possibility of providing efficient packing.

As should any motor, the hydraulic cylinder must have the highest possible efficiency which depends mainly upon the design of the sealing facilities, packing material and its coefficient of friction.

A great many different arrangements and designs of hydraulic cylinders have been developed to comply with various operating conditions and for various capacities. Nevertheless, the most widely employed type in the engineering industries is the asymmetrical hydraulic cylinder (with the rod on one side), as shown in Fig. 87, due to its wide versatility and small size ($L > 2l - b$) as compared to a symmetrical cylinder ($L > 3l - 2b$, see Fig. 88) having a double-end rod.

If oil is delivered at the same rate of flow first to the head (left) end of an asymmetrical (single-end rod) cylinder, and then to the rod (right) end, the velocity of the piston will be different in the two cases since

$$v_1 = \frac{4Q}{\pi D^2} \quad \text{and} \quad v_2 = \frac{4Q}{\pi (D^2 - d^2)}$$

and therefore

$$v_2 > v_1$$

If both ends of the cylinder are connected to the discharge line of the pump (in a differential circuit) the piston will travel rapidly on the forward stroke [(see Fig. 83, Eq. (104) and also Figs. 172, 194, and 201 of Vol. 1)].

The use of double-end rod cylinders, which may involve an increase in the overall dimensions of the machine tool since $L = 3l - 2b$ (see Fig. 88), can be avoided, and the forward and reverse speeds can be made equal in a single-end rod cylinder by designing the cross-sectional area of the rod $\left(\frac{\pi}{4} d^2\right)$ to be one half of the piston area, i.e. so that $d = \frac{D}{\sqrt{2}} \cong 0.71\ D$. This

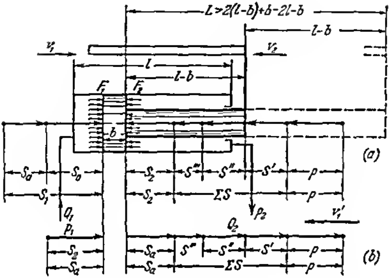

Fig 87 Calculation diagram for a single end rod power cylinder

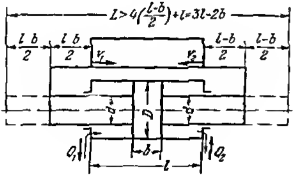

Fig 88 Diagram of a double end rod power cylinder

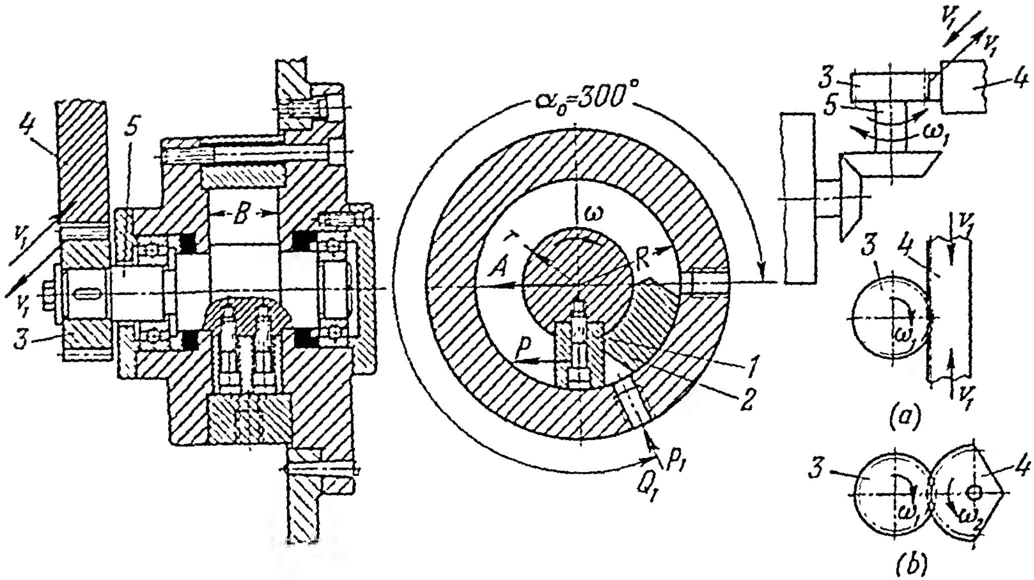

Fig. 89. Hydraulic limited rotary motor:

1—vane; *2*—sector separating the pressure and drain lines; *3*—rack pinion; *4*—table rack; *5*—vane shaft

may be required in such machines as grinders. Then, when the piston travels to the right, both ends of the cylinder are connected together and to the pump so that the delivery into the head end is proportional to $D^2 = 2d^2$. When the piston travels to the left, only the rod end is connected to the pump, the head end drains to the tank and the delivery into the rod end is proportional to $D^2 - d^2 = d^2$.

In a single-acting hydraulic cylinder oil is delivered under pressure only to one end, so that the piston can travel in only one direction under the pressure generated by the pump. The piston must be returned, in this case, by some other source of energy such as a spring, compressed air, etc.

The pistons of double-acting cylinders travel in both directions under the action of the pressure produced by the pump.

The two ends of double-end rod cylinders have the same effective area, i.e. $F_1 = F_2$ (Fig. 88). The effective areas in the ends of single-end rod cylinders (Fig. 87) differ since

$$F_1 = \frac{\pi}{4} D^2 > F_2 = \frac{\pi}{4} (D^2 - d^2)$$

Limited rotary motors, in which the vane has a limited rotary motion in respect to the cylinder (Fig. 89), find extensive application in the feed

mechanisms of machine tools in the handling devices of transfer machines in unit built machine tools and in various auxiliary mechanisms In accordance with the purpose of the motor its output shaft may be linked to the driven mechanism in various ways either through a pinion and rack arrangement (Fig 89a) to provide reciprocating motion, through a pinion and segment gear (Fig 89b) for rotary oscillating motion or through a ratchet mechanism if intermittent motion is required Upon each oscillation through an angle α a vane with the width B displaces a volume of oil equal to $\frac{\alpha}{2}(R^2 - r^2)B$ where α is expressed in radians and R and r are the large and small radii of the cylinder (Fig 89) If the number of vane oscillations is denoted by n the displacement per minute will be

$$Q = \frac{\alpha (R^2 - r^2) nB}{2} \text{ litres per min}$$

In cases when the oscillating cylinder drives the work table of a machine tool through the gearing shown in Fig 89a the table speed is

$$v_1 = \pi m z i n$$

where i = ratio of the bevel gearing
m = module of the rack pinion
z = number of teeth on the rack pinion

Combining the last two equations we obtain

$$n = \frac{2Q}{\alpha (R^2 - r^2) B} = \frac{v_1}{\pi m z i}$$

Hence the required delivery per minute is

$$Q = \frac{\alpha v_1 B}{2\pi m z i}(R^2 - r^2) \tag{111}$$

For a value of $\alpha \simeq 2\pi$

$$Q \simeq \frac{v_1 B}{m z i}(R^2 - r^2) \tag{111'}$$

The torque developed on the shaft of the oscillating motor is

$$M_t = p\frac{B}{2}(R^2 - r^2) \tag{112}$$

and the force exerted by the oil on the vane is

$$P = pB(R - r)$$

It is evident from equation (111) that if a limited rotary motor is used a low delivery pump can provide high table speeds providing that the values m z and i have been properly selected

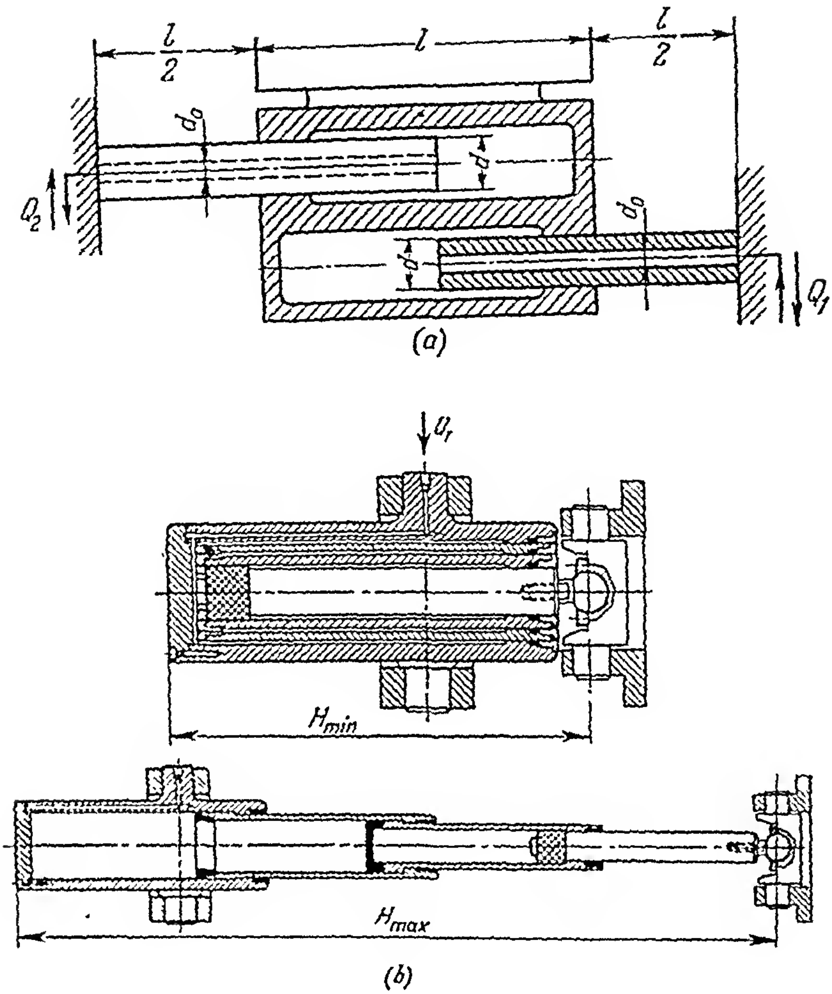

Fig. 90. Plunger systems:
(a) simple system; (b) telescopic system

Because of the limited rigidity of the vanes, limited rotary motors are not used for pressures $p > 10$ to 12 kgf per sq cm.

In machine tools operating with a high pulling force or with a long piston stroke (for example, broaching machines), the piston rod is subject to a tensile load, so that the working stroke in Fig. 87 is from right to left, along arrow v_2.

In plunger systems (Fig. 90) the plunger is usually subject to a buckling load. The axial load can be decreased by making the plunger of hollow

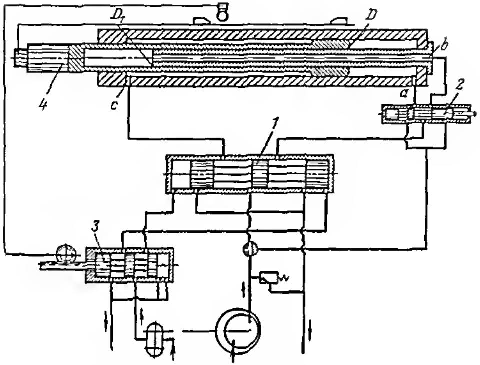

Fig 91 Diagram of a summating hydraulic power cylinder
1—two-position reversing directional valve 2—valve for switching over pump delivery 3—pilot valve for operating the reversing valve 4—piston

design Thus

$$S_1 = s\left(1 - \frac{d_1^2}{d^2}\right)$$

where

$$s = p\frac{\pi}{4}d^2$$

Plunger systems are employed in telescopic hydraulic cylinders (Fig 90*b*) in which the total stroke H_{max} of all the plungers can be substantially longer than the length H_{min} of the cylinder Such systems are characterized by the number of stages (from two to six), each stage having a stroke up to 1,500 mm Summating hydraulic cylinders (Fig 91) are frequently used to vary piston speeds in a stepped range or to obtain several different pulling forces * When the discharge Q of the pump is delivered simultaneously to

* See also Fig 183 (Vol 1) in which another type of summating cylinder is employed

both right-hand ports of the cylinder, piston *4* travels at its lowest speed

$$v_{1min}=\frac{4Q}{\pi(D^2-D_1^2)+\pi D_1^2}=\frac{4Q}{\pi D^2}$$

The middle speed of the piston is

$$v_1=\frac{4Q}{\pi(D^2-D_1^2)}$$

and it is obtained by delivering the discharge of the pump to port *a*, port *b* being connected to the tank. The maximum speed of the piston is

$$v_{1max}=\frac{4Q}{\pi D_1^2}$$

It is obtained by delivery to port *b* with port *a* connected to the tank.

The pulling forces for each step vary inversely proportional to the piston speeds. Thus, $P_{max}=p\frac{\pi D^2}{4}$; $P=p\frac{\pi}{4}(D^2-D_1^2)$ and $P_{min}=p\frac{\pi D_1^2}{4}$. Return strokes are obtained by delivering oil to port *c*.

If the pressure developed by the pump is insufficient, use is made of a hydraulic intensifier, or booster which enables the pressure to be increased to the required value (Fig. 92).

Cylinder *1* and upright *3* of the intensifier are rigidly mounted in the bed of the machine. The movable member is plunger *2*. In charging the intensifier, the spool of directional valve *9* is shifted to the left. At this, plunger *2* begins to travel upwards since the pump delivers oil through check valve *4* and upright *3* into the cavity of plunger *2*. At the same time, oil is admitted into the rod end of hydraulic cylinder *5* to shift its piston to the initial position. In this position, the system is charged and ready for the working stroke.

The working stroke is started by shifting the spool of valve *9* to the right. This connects discharge line p_p to the upper end of intensifier cylinder *1*; the pressure forces plunger *2* downwards delivering oil at a pressure p_1 to the head end of cylinder *5*.

From equilibrium conditions for plunger *2* it follows that

$$p_pF_1+G-\sum S=p_1f$$

Therefore the pressure in the hydraulic cylinder is

$$p_1=p_p\frac{F_1}{f}+\frac{G-\sum S}{f} \tag{113}$$

where $F_1=\frac{\pi D_1^2}{4}$ = cross-sectional area of the plunger

$f=\frac{\pi d_1^2}{4}$ = cross-sectional area of the upright

G = weight of the plunger

$\sum S$ = total friction of the packing in the cylinder and plunger of the intensifier.

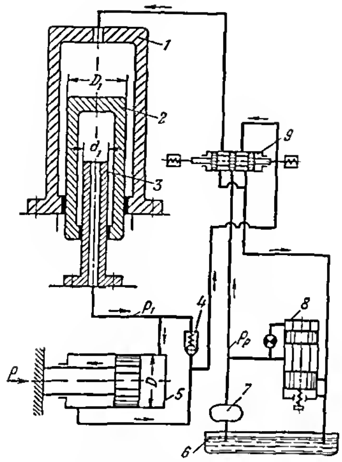

Fig 92 Intensifier circuit diagram

1—intensifier cylinder 2—intensifier plunger 3—upright 4—check valve, model Г51 5—power cylinder 6—tank 7—pump 8—backpressure valve model Г54 9—directional valve

In cases in which the piston speed must be varied in a large range, for example 200 : 1, and the stroke is very long, the rigidity of the hydraulic system is considerably reduced due to the large capacity. The need for such a long cylinder can be avoided by employing a rotary hydraulic motor, for example model МГ-15 (USSR), linking its output shaft by suitable toothed gearing to the rack of the work table

Rotary hydraulic motors must have a sufficiently high volumetric efficiency so that variations in load do not appreciably affect the speed of the machine tool unit, for example work table, that is being driven

By a corresponding selection of the gearing ratio between the output shaft of the rotary motor and the table rack, it is possible to obtain very small movements of the table, much smaller than would be possible using a hydraulic cylinder.

The minimum speeds and the range of speed variation with the output flow passing through a metering valve (metering-out arrangement) are given in Table 3 for rotary hydraulic motors of the МГ-15 series.

TABLE 3

Item	Model				
	МГ 151	МГ 152	МГ 153а	МГ 154а	МГ 155а
Minimum speed, rpm	$\frac{16^*}{40}$	$\frac{8^*}{30}$	$\frac{4^*}{20}$	$\frac{2^*}{20}$	$\frac{1^*}{20}$
Range of motor speed variation	150 : 1*	260 : 1*	450 : 1*	650 : 1*	1,300 : 1*

* With metering-out controls.

If this type of axial-piston motor is arranged to operate in conjunction with a self-priming pump (see Fig. 82), they can operate as pumps which increase the pressure during the working stroke.

The motor housing *10* (Fig. 93) contains cylinder barrel *1* with pistons *2*, driving disk *3* with tappets *4* and drive shaft *7*. Disk *3* is keyed on shaft *7*. Cylinder barrel *1*, mounted freely on the same shaft and located by a narrow band, is driven by disk *3* only through driving stud *8*.

The radial ball bearing of shaft *7* is mounted in cover *5*; the other end of the shaft is supported by a bearing in the distributing disk. This disk has channels for connection to the pressure and return lines. The disk has four ports separated by partitions. Two serve for suction and two for discharge. Cylinder barrel *1* is connected through these ports with the suction and discharge lines.

The oil pressure developed by the pump acts on pistons *2* which push tappets *4* forward, up against ball thrust bearing *6*. This bearing is mounted in cover *5* at a definite angle to drive shaft *7*.

Tangential forces, obtained by resolving the normal forces and the forces of the pressure on the pistons, rotate driving disk *3* by means of tappets *4*. Through stud *8*, disk *3* drives cylinder barrel *1*. Thus the tappets transmit only axial forces to the driving disk, while the tangential forces are carried by the tappets, for which purpose they have ample bearing surfaces in driving disk *3*.

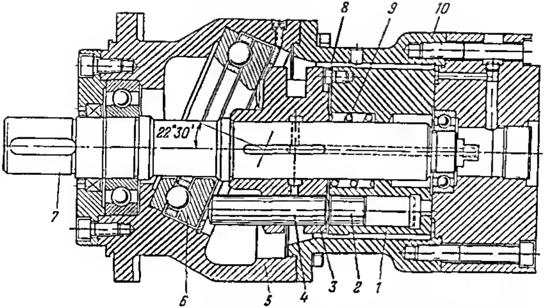

Fig 93 Hydraulic axial piston motor, model MΓ15, designed by ENIMS·
1—cylinder barrel 2—piston 3—driving disk, 4—tappet 5—cover, 6—ball thrust bearing 7—drive shaft, 8—driving stud 9—spring for holding the barrel against the distributing disk, 10—motor housing

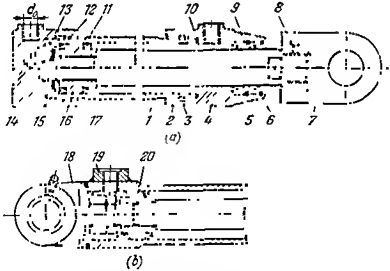

Fig 94 Power cylinder construction
1—cylinder tube 2—bracket 3—split fastener ring 4—front cover 5—packing cup 6—snap ring for holding the packing cup 7—interchangeable rod end 8—set screw for locking the rod end 9—intermediate washer of the packing gland 10—sealing ring 11—piston sealing cup 12—lock washer 13—nut, 14—rear cover 15—washer 16—piston 17—rod 18—clevis 19—boss 20—packing seal

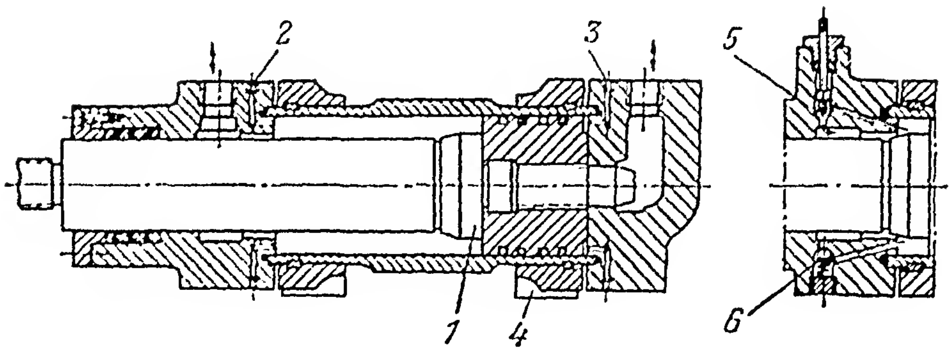

Fig. 95. Cushioned power cylinder:

1—cushioning ring; 2 and 3—air-bleeding plugs; 4—bracket; 5—throttle valve; 6—check valve

Hydraulic cylinders are usually assembled units consisting of cylinder tube *1* (Fig. 94*a*), brackets *2* for mounting the cylinder on the machine, covers *14* and *4* with holes d_0 for connection to the pressure and exhaust lines, rod *17* and packing *5*.

In pendulum cylinders (Fig. 94*b*), the brackets are replaced by clevis *18*.

Brackets *2* are fastened to cylinder tube *1* with half rings *3*. Such a construction facilitates manufacture. Annular grooves are turned for this purpose in the tube to hold the two half-rings.

Hydraulic cylinders designed with a cushioning device for reversals have a somewhat longer seating surface on the rod for the piston (Fig. 95). Tapered ring *1* is fitted on the packing side of the rod behind the piston while a plain cylindrical part of the rod projects from the opposite end of the piston.

The escaping fluid is forced to pass through choke or needle valves *5*, mounted in the two heads and used to regulate the cushioning effect. Check valves *6* mounted in the heads permit oil to enter for the return stroke.

After ring *1* enters the closely fitted counterbore in the head, the oil can escape only through needle valve *5*. By adjusting this valve, the deceleration and acceleration times of the piston can be varied. Plugs *2* and *3* enable air to be bled from the cylinder.

Cylinder tubes for machine tool hydraulic systems are made of steel tubing (USSR Std GOST 8732-58). Overall dimensions of Soviet-made cylinders, their parts and the rules for their installation and for hydrostatic pressure testing are stipulated by ENIMS standards Г21-10 and Г22-10.

The following diameters of hydraulic cylinders and the corresponding pump discharges are recommended for general practice (Table 4).

The minimum pressure p_{min} characterizes the friction losses in the rod and piston packing.

TABLE 4

Cylinder diameter D mm	Pump discharge Q litres/min	Minimum pressure p_{min} in system kgf/cm²	Permissible volumetric losses q at maximum pressure, cm³/min
45, 55	35	4	4 to 5
65, 75, 90	70	2 5 to 3 5	9 6
105, 125	100	1 5 to 2	10 to 12
150, 180	140	0 5 to 1	15 to 18

The choice of the type and diameter of a hydraulic cylinder depends upon the operating cycle of the machine, i e upon the working and rapid traverse motions and the required force

All hydraulically operated machine tools can be classified tentatively into three groups in accordance with the piston speed and force required

A Group of grinders and honing machines operating with small forces and high piston speeds, pressure ranges up to 20 kgf per sq cm and the power rating is up to 4 5 or 5 kW

B Group of planers slotters and broaching machines operating at high pressure (up to 70 or 75 kgf per sq cm), mean piston speeds range up to 30 or 35 m per min and the power rating is up to 50 or 60 kW

C Group of lathes boring, drilling and milling machines operating with large forces and low working speeds (up to $v_1 \cong 0\ 5$ m per min) the pressure in the system may range up to 60 or 65 kgf per sq cm and the power rating up to 3 or 4 kW

The diameter of the cylinder is selected in a different way for each group Thus, for machine tools of group A, this diameter is calculated on the basis of the given ratio of forward and return speeds, v_1 and v_2, respectively In this way

$$k = \frac{d}{D} = \sqrt{1 - \frac{v_1}{v_2}} \tag{114}$$

If approximately equal forward and return speeds are required, the rod diameter should be $d = (0\ 2$ to $0\ 3)\ D$

In designing machine tools of groups B and C the cylinder diameter is chosen on the basis of the given useful load P_u after first selecting the pressure p in the cylinder in accordance with this load

P_u tons =	1 to 2	1 to 3	3 to 5	5 to 10
kgf/cm² =	15	35	50	65

Thus, if the useful load P_u is given, and the pressure p in kgf per sq cm in the cylinder has been assigned, the piston diameter can be readily determined from the equation

$$D = 2\sqrt{\frac{P_u}{\pi p}} \tag{115}$$

The rod diameter for these groups is usually within the limits $d = (0.5$ to $0.7)\ D$. The final values of the cylinder and rod diameters are corrected in Soviet practice to those listed in USSR Std GOST 6540-53 and ENIMS standard H21-3.

Taking into consideration the cylinder manufacturing processes, requirements made to hydraulic system rigidity, installation conditions and other factors, it is advisable to keep the ratio of the cylinder length l to its diameter D at values less than 20.

11-2. Packings and Seals

Packings and seals are used for both movable and fixed joints. In either case, they may be of adjustable or self-tightening design.

In stationary applications, packings and seals must ensure airtightness in the whole range of working pressures and temperatures; they must be convenient in assembly and readily disassembled.

The degree of airtightness in movable joints is characterized by the amount of working fluid that leaks through the packing in unit time. Therefore, the maximum leakage is limited by standards and depends upon the type and material of the packing or seal, and the purpose of the hydraulic device in which it is to be installed.

In adjustable packing the sealing effect is produced by tightening an adapter; the tightening force may vary in a wide range and depends upon the skill of the fitter that is making the adjustment.

Fixed joints that are tightened by bolts may be of the plain or gasket type. The sealing effect is produced in plain joints (ones without gaskets) by machining the contacting surfaces to a sufficiently high class of finish (10th or 11th according to USSR Std GOST 2789-59). The second vital factor influencing airtightness is the tendency of the contacting surfaces to wetting. Experience shows that it is easier to seal unwettable surfaces, especially if they are coated with a thin layer of fat or oil.

If the contacting surfaces are only ground, a gasket is used to improve the tightness of the joint. The radial forces tending to force the gasket out of the joint increase with the thickness of the gasket. Thin gaskets, such as tracing paper coated with fat or oil, are used to reduce these radial forces.

In self-tightening movable and stationary seals and packings, the pressure of the fluid promotes the sealing effect. The higher the fluid pressure, the

better the airtightness of the packing (or seal) A necessary condition for reliability and airtightness of self tightening packing is initial compression or preloading produced by squeezing the packing with some tightening facilities

U cups are widely used for stationary self tightening packing applications (Fig 96a and b) These cups are homogeneous in design and are made in moulds from oil resistant rubber grades 3825 and 4004 according to Specifications 1166 58 of the USSR Ministry of the Chemical Industries

To install the U cup recesses 2 to 2 5 mm deep are made in the contacting face of each of the parts to be joined The U cup *3* is installed with a slight interference in the groove formed by the recesses and the bolts are drawn up to fasten the two parts The pressure of the fluid inside the cup holds its sealing lips against the walls of the recess with a force proportional to the pressure

U cup packing is especially convenient in connecting piping to a hydraulic device pump etc (Fig 96b)

The material of the packing must meet certain definite requirements it must retain its strength under various types of loads it must withstand the pressure and temperature of the hydraulic fluid it must not absorb the fluid nor decompose or be oxidized by the action of the fluid

Soft packing of plastics based on oil resistant rubber (Fig 97a) is widely employed for sealing reciprocating components The number of cups in a stack (Fig 97b and c) depends upon the sliding speed and the fluid pressure At high sliding speeds (as in grinders for example) no more than two cups are used so as to reduce the friction forces and the rate of wear More cups are employed for higher pressures and lower sliding speeds

According to ENIMS standard A58 1 the oil resistant rubber of cups type 3825 from $d = 12$ mm and up can be replaced by cotton cloth (USSR Std COST 6i2 41) *impregnated with an oil resistant and graphitized compound*

The force with which the cups are compressed can be adjusted by the use of a set of shims *1* (Fig 97c) made of 1 mm sheet steel The shims are removed as required and the adapter is drawn up tight

Another type of U cup packing with a thicker body (Fig 97d) is finding widespread application for sealing pistons and rods These cups are preloaded by the elasticity of the cup material and by properly selecting the angle at which the sides are inclined These cups are noted for their low friction and comparatively high dependability

O ring and quad ring packings and gaskets (Fig 97e and f) made of the same oil resistant rubber grade 3825 as mentioned above provide excellent service in stationary separable and movable applications Such rings are highly resilient and have a large coefficient of compressibility O ring can be installed in half round (Fig 98a) or vee (Fig 98b) grooves Rectangular

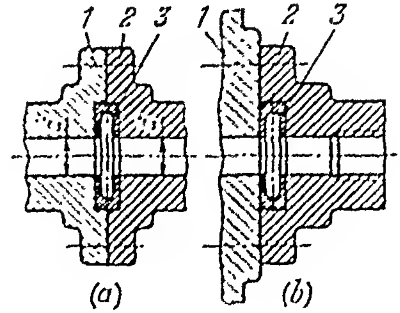

Fig. 96. U-cup packing for sealing stationary joints:
1 and 2—components being sealed; 3—U-cups

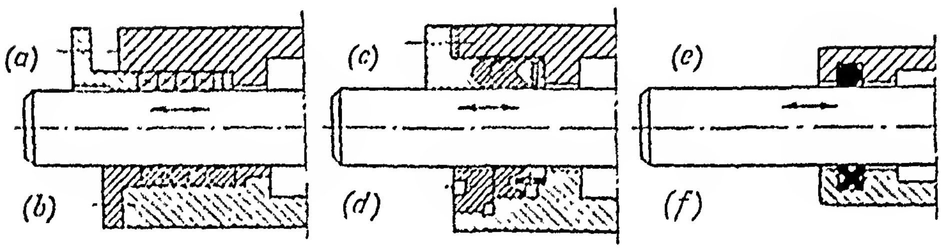

Fig. 97. Soft packing for sealing reciprocating components:
(a) ordinary soft packing; (b) V-ring packing; (c) stacked U-cups; (d) single U-cup with support ring; (e) O-ring packing; (f) quad-ring packing

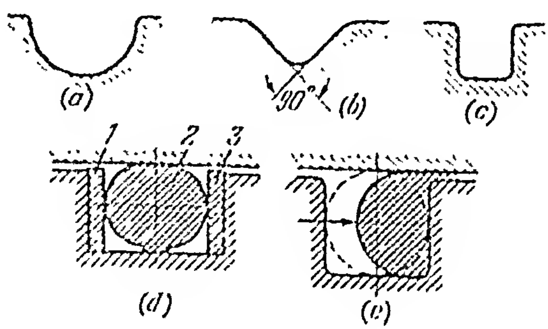

Fig. 98. Grooves for O-ring packing

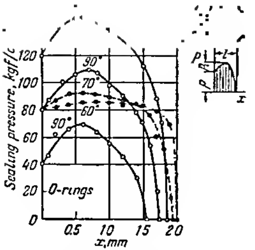

Fig. 99. Pressure variation of O rings of various hardness with $d = 2.6$ mm and $\frac{\Delta d}{d} = 0.12$, along their length of contact

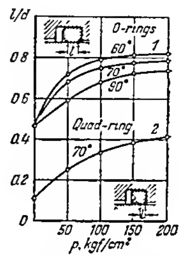

Fig. 100. Resilience $\frac{l}{d}$ of O rings (1) and quad rings (2) of various hardness vs the pressure of the medium being sealed in. O rings have hardnesses of 70, 80 and 90, the quad-ring hardness is 70, $\frac{\Delta d}{d} = 0.12$

grooves (Fig. 98c) are used for both O rings and quad-rings. Vee and half round grooves are applicable for pressures up to $p = 70$ kgf per sq cm; ring can be easily installed in them or removed. It is advisable to use rectangular grooves for higher pressures (up to $p = 120$ kgf per sq cm); they can be used for both stationary and movable applications.

At high and extra high pressures, up to $p = 600$ kgf per sq cm, rings 2 are installed together with two nonextrusion rings 1 and 3 (Fig. 98d). Under pressure the ring is shifted in the direction the pressure acts. It is deformed and, as a result, makes tight contact with the surfaces to be sealed. The rubber acts as would a viscous fluid, transmitting pressure in all directions.

It is evident from the pressure-distribution curves for O rings (Fig. 99) that the pressure is not uniform along the length of contact l of the ring. Of great importance is the hardness of the ring. Thus, for example, for a ring with a Shore hardness of 90 the point with the maximum sealing pressure is located at a distance of 0.5 to 0.6 mm from the origin of the co-ordinate system (point where the fluid pressure is applied). The sealing pressure then drops and is almost equal to zero at a distance of 2 mm. In soft rings (Shore hardness 60 to 70), the pressure of the ring almost coincides with the fluid pressure p over the whole length l of ring contact.

The ratio of the seal length l to the ring diameter d indicates the resilience of the cross-sectional shape of the ring.

The curves of Fig. 100 show that the hardness of the ring material has no appreciable effect on the resilience. On the other hand, the shape of the cross section does affect the resilience. For example, the resilience of a quad-ring is only about one half of that of an O-ring of the same hardness.

The shape of the ring also affects its sealing properties. Thus even though a quad-ring is less resilient than an O-ring, it seals a piston better at the same fluid pressure (Fig. 101).

The application of plastics for packing cups enables them to be manufactured with the most rational shapes, and permits the preload to be varied. By changing the components of the plastic it is possible to produce a packing that is more suitable to the working conditions, to vary the coefficient of friction, etc. This cannot be done if the packing is made of materials of animal or vegetable origin (leather, hemp, etc.). The hardness, for instance, of the plastic ring or cup is varied by changing the sulphur content; zinc oxide is added to the mixture to provide resilience; the addition of soot and graphite changes the heat conduction, etc.

A plastic packing fits tightly to the surface being sealed. Hence, this surface must be machined to a high mirror finish to avoid premature wear of the packing; annular scratches and tool marks are especially harmful in this respect. The height of the irregularities on the surface being sealed must not exceed 0.4-0.8 microns.

The use of cast-iron split piston rings (made of grade СЧ 21-40 C.I. according to USSR Std GOST 1412-54) is a universal method of sealing pistons operating with various types of fluids at pressures up to 500 kgf per sq cm. The rings are initially held against the wall of the hydraulic cylinder (Fig. 102) by their internal elastic forces σ. During operation, the pressure of the fluid in the cylinder additionally forces the ring against the cylinder wall over the length b (Fig. 102). The first ring is subject to the maximum pressure drop $\Delta p' = (0.75 \text{ to } 0.8)\, p$. If the pressure drop is excessively large, i.e. $\Delta p > 250$ kgf per sq cm, the ring is forced against the groove in the piston. As a result, the piston ring is jammed and loses its sealing properties.

It has been proved in practice that the service life of metallic piston rings is many times longer than that of plastic packings.

In the Soviet Union, piston rings for hydraulic pistons from 30 to 1,000 mm in diameter have been standardized by ENIMS Std A54-1.

The use of pistons without any packing or seal is based on precise lapping-in of the surfaces to be sealed to obtain micron clearances. There is very little friction if the piston and cylinder tube are carefully machined and assembled, and good leakproof properties are obtained if a concentric (symmetrical) diametral clearance is maintained between the piston and bore. This arrangement is highly sensitive to changes in temperature of the fluid, and seizing

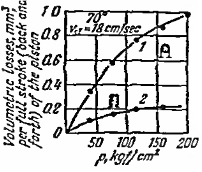

Fig. 101 Volumetric losses in the power cylinder using O ring (1) and quad ring (2) packing vs the pressure of the medium Shore hardness of the rings is 70, sliding velocity — 18 cm per sec

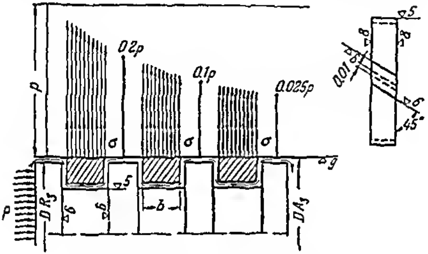

Fig 102 Sealing with metallic piston rings

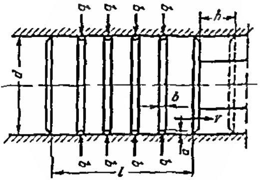

Fig 103 Piston without packing

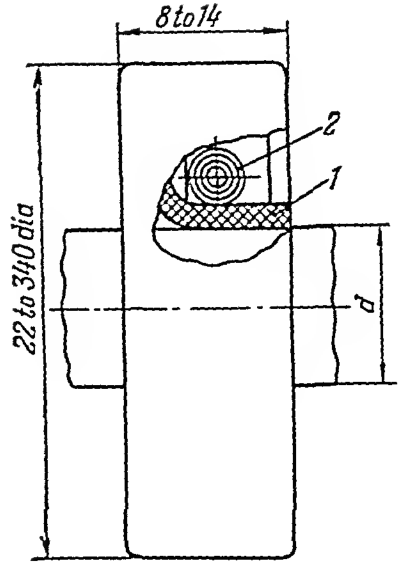

Fig. 104. Press-fit seal for a rotating shaft

of the piston may occur if the allowance is too small. An eccentric diametral clearance leads to the development of unilateral forces that may also cause the piston to jam. Fluid leakage is increased in all cases of eccentric clearances.

The provision of annular grooves $a = 0.3$ mm deep and $b = 0.5$ mm wide (Fig. 103) floats the piston, relieving it of unilateral forces.

Of vital importance in reducing the volumetric losses of such pistons and cylinders is a proper selection, in accordance with the fluid pressure, of the ratio $\frac{l}{d}$, where l is the length of the sealing surface and d is its diameter. This sealing arrangement can be used for pressures up to $p = 2{,}000$ kgf per sq cm, and the ratio $\frac{l}{d}$ ranges from 0.75 to 15. Pistons without packing are used mainly in hydraulic controls (valves, relays, etc.).

Plastic packings are used at relative velocities up to 1.2 m per sec, while sealing by lapping-in procedures can be used for relative velocities up to 3 m per sec.

Press-fit seals for rotating shafts of a diameter $d = (6 \text{ to } 300)$ mm are listed in ENIMS Std A51-4 (Fig. 104). Here the sealing element is cup *1* of oil-resistant rubber, grade 3825 or 4004 according to Specifications 1166-51 MCI.

If the preload is applied only by annular spring *2*, the peripheral speed of the shaft may reach 12 m per sec. If, in addition to spring *2*, a fluid pressure of $p = 1$ kgf per sq cm acts on the cup, the peripheral speed should not exceed 8 m per sec. Substituting leather for the oil-resistant rubber enables the permissible peripheral speed to be increased to 10 m per sec. This is evidently due to the porous structure of the leather (the pores are filled with lubricant).

Friction losses in plastic packings and seals depend upon the pressure of the fluid being sealed off, the area of the sealing surface, the coefficient of friction between the packing and surface being sealed, and the preload. The friction force is

$$S = S_0 + pfF \text{ kgf} \tag{116}$$

where S_0 = preload of the cup or ring, kgf

f = 0.10 to 0.17 = coefficient of friction

p = fluid pressure, kgf per sq cm

F = contacting surface of the packing, sq cm.

If $\frac{l}{d}$ is denoted by φ, in O ring packing, then

$$F = \pi l d = \pi \varphi d^2$$

The friction force developed by metallic piston rings (see Fig 102) is

$$S = f p_r F \text{ kgf} \tag{117}$$

where p_r is the pressure exerted by the rings on the cylinder wall, kgf per sq cm

$$p_r = \left(\sum p_r + z\sigma\right) \text{ kgf per sq cm}$$

where $\sum p_r$ = sum of the pressures of the rings, kgf per sq cm (for three rings $\sum p_r = 0.83p$)
z = number of rings (usually $z = 3$ or 4)
σ = 0.5 to 0.6 kgf per sq cm = preload pressure

11-3. Statics and Dynamics of the Hydraulic Cylinder

The compressibility of oils is negligible and therefore their pressure is transmitted completely to the piston, minus the backpressure and friction losses. Work transmitted by the fluid is proportional, all other conditions being equal, to the rate of flow, while the pressure loss in the system is proportional to the square of the velocity.

In its motion the piston must overcome (see Fig 87) the resistance of the useful load P, sum of the friction forces $\sum S$, force S_2 exerted by the backpressure and the dynamic force S_a which is required to reach the running speed in a given interval of time or, in other words, the specified acceleration.

The following sum of forces are usually referred to as the *static forces* acting on the piston

$$S_0 = R + S_2 + T$$

where $R = P + S$, S being the friction forces in the work table ways
$T = S' + S''$ sum of the friction forces in the rod packing S' and in the piston packing S''

Hence, the force acting on the piston during acceleration is

$$S_1 = S_0 + S_a$$

while during steady travel of the table it is

$$S_1 = S_0$$

When the piston diameter has been selected or given, the required pressure of the oil entering the cylinder is determined from the magnitude of force S_1 for the acceleration period.

If the working stroke of the piston is selected in direction v_1 (see Fig. 87*a*), then the force $R = P + S'$ is applied to the rod, subjecting it to a buckling load, and the force $T = S'' + S'''$ acts on the piston.

If the working speed is in the direction v_1' (see Fig. 87*b*) as, for example, in broaching machines, the rod is subject to a tensile load which is

$$T' = P + \sum S + S_a$$

The backpressure S_2, in this case, acts on the piston. The pressure p_p with which the fluid should be discharged from the pump is

$$p_p = p_1 + \Delta p_i$$

where p_1 = pressure in the hydraulic cylinder

Δp_i = loss of pressure in the pressure and exhaust lines.

Making use of Bernoulli's equation

$$p_p + \gamma \frac{v^2}{2g} = p_1 + \gamma \frac{v_1^2}{2g} + \Delta p_i \tag{118}$$

and the equation of continuity

$$vF_0 = v_1 F_1$$

where v_1 = piston velocity

v = velocity of the fluid in the pressure line

F_1 = cross-sectional area of the cylinder

F_0 = internal cross-sectional area of the pressure line piping,

we can write

$$p_p - p_1 = \gamma \frac{v_1^2}{2g}\left(1 - \frac{F_1^2}{F_0^2}\right) + \Delta p_i \tag{119}$$

Since $\frac{F_1^2}{F_0^2} \gg 1$, then $1 - \frac{F_1^2}{F_0^2} \cong -\frac{F_1^2}{F_0^2}$. In this case, $F_0 = \frac{\pi d_0^2}{4}$, where d_0 is the internal diameter of the pressure line piping. Only pressure losses from local resistances are taken into consideration. Then

$$p_p = p_1 - 1.6\gamma \frac{v_1^2 F_1^2}{2g d_0^4} + 1.6\gamma K_0 \frac{Q^2}{2g d_0^4}$$

and since $K_0 \gg 1$ in equation (81)

$$p_p = p_1 + 1.6\gamma K_0 \frac{v_1^2 F_1^2}{2g d_0^4} \text{ kgf per sq cm} \tag{120}$$

in which γ is expressed in kgf per cu cm, v_1 in cm per sec, F_1 in sq cm, g in cm per sec^2 and d_0 in cm.

If we express the forces acting on the piston by means of the corresponding areas and pressures for the conditions of steady motion, i.e. for $S_1 = S_0$, then

$$p_1F_1 = pF_1 + p_SF_1 + p_2F_2$$

where p = pressure to overcome the load
p_S = pressure to overcome the friction forces
p_2 = backpressure

and the pressure in the cylinder is

$$p_1 = p + p_S + p_2\frac{F_2}{F_1} \quad (121)$$

The power transmitted to the hydraulic cylinder is

$$N_1 = \frac{v_1F_1}{1.02 \times 10^3}(p_p - \Delta p_i) \text{ kW} \quad (122)$$

The indicated power of a hydraulic cylinder is

$$N_i = N_1 - N_S = p\frac{v_1F_1}{1.02 \times 10^3} \text{ kW} \quad (123)$$

where N_S = power lost in overcoming the friction forces and the backpressure.

Thus

$$N_S = \frac{v_1F_1}{1.02 \times 10^3}\left(p_S + p_2\frac{F_2}{F_1}\right) \quad (124)$$

The mechanical efficiency of a hydraulic cylinder is

$$\eta_m = 1 - \frac{N_S}{N_1} = 1 - \frac{p_S + p_2\frac{F_2}{F_1}}{p_p - \Delta p_i} \quad (125)$$

If the backpressure is specially provided for, as in systems with stabilized speed, its value may vary in a fairly wide range, p_2 = 4 to 15 kgf per sq cm. In all other cases, p_2 = 0.5 to 1.5 kgf per sq cm, and can therefore be neglected.

A hydraulic piston has one degree of freedom. Therefore, the position of the piston is characterized by co-ordinate x.

Using d'Alembert's equation, we can write

$$M_0\frac{d^2x}{dt^2} = \Theta \quad (126)$$

where Θ is the summated force, acting on the piston of the hydraulic cylinder

When the piston diameter has been selected or given, the required pressure of the oil entering the cylinder is determined from the magnitude of force S_1 for the acceleration period.

If the working stroke of the piston is selected in direction v_1 (see Fig. 87*a*), then the force $R = P + S'$ is applied to the rod, subjecting it to a buckling load, and the force $T = S'' + S'''$ acts on the piston.

If the working speed is in the direction v_1' (see Fig. 87*b*) as, for example, in broaching machines, the rod is subject to a tensile load which is

$$T' = P + \sum S + S_a$$

The backpressure S_a, in this case, acts on the piston. The pressure p_p with which the fluid should be discharged from the pump is

$$p_p = p_1 + \Delta p_l$$

where p_1 = pressure in the hydraulic cylinder

Δp_l = loss of pressure in the pressure and exhaust lines.

Making use of Bernoulli's equation

$$p_p + \gamma \frac{v^2}{2g} = p_1 + \gamma \frac{v_1^2}{2g} + \Delta p_l \tag{118}$$

and the equation of continuity

$$vF_0 = v_1 F_1$$

where v_1 = piston velocity

v = velocity of the fluid in the pressure line

F_1 = cross-sectional area of the cylinder

F_0 = internal cross-sectional area of the pressure line piping,

we can write

$$p_p = p_1 + \gamma \frac{v_1^2}{2g}\left(1 - \frac{F_1^2}{F_0^2}\right) + \Delta p_l \tag{119}$$

Since $\frac{F_1^2}{F_0^2} \gg 1$, then $1 - \frac{F_1^2}{F_0^2} \approx -\frac{F_1^2}{F_0^2}$. In this case, $F_0 = \frac{\pi d_0^2}{4}$, where d_0 is the internal diameter of the pressure line piping. Only pressure losses from local resistances are taken into consideration. Then

$$p_p = p_1 - 1.6\gamma \frac{v_1^2 F_1^2}{2g d_0^4} + 1.6\gamma K_0 \frac{Q^2}{2g d_0^4}$$

and since $K_0 > 1$ in equation (81)

$$p_p \approx p_1 + 1.6\gamma K_0 \frac{v_1^2 F_1^2}{2g d_0^4} \text{ kgf per sq cm} \tag{120}$$

in which γ is expressed in kgf per cu cm, v_1 in cm per sec, F_1 in sq cm, g in cm per sec² and d_0 in cm.

If we express the forces acting on the piston by means of the corresponding areas and pressures for the conditions of steady motion, i.e. for $S_1 = S_0$, then

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where Θ is the summated force, acting on the piston of the hydraulic cylinder.

In the given case (see Fig. 87) this force equals

$$\theta = F_1\left(p_p - cv_1^2 - p_S - p_2\frac{F_2}{F_1}\right) \tag{127}$$

M_0 is the referred mass of parts in translatory motion

$$M_0 = \sum m_j \frac{v^2}{v_1^2} \tag{128}$$

where v_1 — translational velocity of the piston
v — oil velocity in the pressure line.

After making the substitution $\frac{v}{v_1} = \frac{F_1}{F_0}$, the referred mass is

$$M_0 = M_1 + M_2\left(\frac{F_1}{F_0}\right)^2 + M_3\left(\frac{F_2}{F_0}\right)^2 \tag{129}$$

where M_1 — mass of parts travelling with translatory motion (piston, work table, etc.)
M_2 — mass of the oil in the pressure line
M_3 — mass of the oil in the exhaust line
F_1 and F_2 — areas of the piston (see Fig. 87)
F_0 — cross-sectional area of the piping.

Thus equation (126) can be written as

$$M_0\frac{d^2x}{dt^2} = F_1\left(p_p - p_S - p_2\frac{F_2}{F_1}\right) - cF_1v_1^2$$

It is assumed that the pump pressure p_p, friction resistance p_S and the force of the backpressure p_2 are constant.

Then the following values are also constant

$$F_1\left(p_p - p_S - p_2\frac{F_2}{F_1}\right) = B_1 \text{ kgf}$$

$$cF_1 = B_2 \text{ kgf-sec}^2 \text{ per sq cm}$$

With these assumptions we obtain

$$M_0\frac{d^2x}{dt^2} = B_1 - B_2v_1^2 \tag{130}$$

Dividing both sides of this equation by M_0, we can write

$$\frac{d^2x}{dt^2} + q\left(\frac{dx}{dt}\right)^2 - r = 0 \tag{131}$$

where $q = \frac{B_2}{M_0}$ (cm^{-1}) and characterizes the piping resistance
$r = \frac{B_1}{M_0}$ (cm per sec^2) and characterizes the level of the initial acceleration of the piston.

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$r = \frac{B_1}{M_0}$ (cm per sec^2) and characterizes the level of the initial acceleration of the piston.

If the time for acceleration is to be determined, the last equation is solved for t. Thus

$$t = \frac{1}{2m}\ln\frac{m+qz_1}{m-qz_1}$$

where $m = \sqrt{rq}$.

On the other hand, if the path length, velocity and acceleration of the piston are to be determined, then equation (131) is solved in respect to the path length x of the acceleration period. Then, making the substitution $\frac{dx}{dt} = z$, and separating the variables we obtain

$$dt = \frac{dz}{r - qz^2}$$

from which

$$\frac{m+qz}{m-qz} = Ce^{2mt}$$

and therefore

$$z = \frac{mCe^{2mt} - 1}{qCe^{2mt} + 1}$$

Substituting for z and integrating we can write

$$x = \frac{1}{2q}\ln\frac{(Ce^{2mt}+1)^2}{Ce^{2mt}} + C_1$$

The constants are determined on the basis of the initial conditions. At $t = 0$, both $z = 0$ and $x = 0$. Hence

$$x = \frac{0.5}{q}\ln\cosh^2(mt) \tag{132}$$

The piston velocity is

$$v = \frac{dx}{dt}\sqrt{\frac{r}{q}}\tanh(mt) \tag{133}$$

The acceleration of the piston is

$$a = \frac{d^2x}{dt^2} = \frac{r}{\cosh^2(mt)} \tag{134}$$

In tentative calculations, sufficiently accurate for practical purposes, we develop a series of hyperbolic functions to find the displacement and velocity of the piston during the period of acceleration. Thus

$$x \simeq 0.5rt^2 \tag{135}$$

$$v \simeq rt \tag{136}$$

If the referred mass M_0 is small in comparison with other members of equation (130), we can assume that $M_0 \frac{d^2x}{dt^2} = 0$ and obtain the piston velocity for the period of steady motion as

$$v_1 = \sqrt{\frac{B_1}{B_2}} \tag{137}$$

Equating the right-hand sides of equations (136) and (137) we can write

$$rT_0 = \sqrt{\frac{B_1}{B_2}} \quad \text{or} \quad \frac{B_1}{M_0} T_0 = \sqrt{\frac{B_1}{B_2}}$$

which we solve for the coefficient T_0 which characterizes the inertia of the hydraulic system. Thus

$$T_0 = M_0 \sqrt{\frac{1}{B_1 B_2}} \text{ sec} \tag{138}$$

If the acceleration a is known (see equation 134), then the dynamic force (see p. 263) is

$$S_a = M_0 a = \frac{B_1}{\cosh^2 (mt)} \tag{139}$$

The acceleration of the piston can also be determined from the acceleration curve, if the slope β of the characteristic curve is given. Thus

$$a = \tan \beta = \frac{v_1}{t}$$

where t is the specified time for acceleration of the piston.

11-4. Cushioning Hydraulic Cylinders

Braking devices are known as *cushioning* facilities if their purpose is to absorb kinetic energy, for instance, at the moment of piston reversal. If the braking device operates during load action, it is called a *damping* arrangement and serves to quench a pulsating load, for example as in valves.

Cushioning is usually accomplished in machine tools by the use of an annular clearance (see Fig. 95) or a special valve whose operative components are profiled in accordance with the nature of the cushioning that is required. An example is the model F31 control panel for grinders (see p. 339). Neglecting the resilience of the hydraulic system, we can write the equation of equilibrium in the cushioning period as

$$T = \left(M_0 \frac{d^2x}{dt^2} - \sum S \right) \text{ kgf} \tag{140}$$

where M_0 = mass to be braked referred to the piston (see p. 266)
$\sum S$ = friction force of the piston (see p. 263) kgf
T = force of resistance of the cushioning aperture

$$T = F\gamma \frac{v_1^2}{2g}(1+\xi) \text{ kgf} \tag{141}$$

where F = piston area on the backpressure side, sq m
v_l = velocity of oil flow in the cushioning bore, m per sec
ξ = coefficient of resistance for the cushioning bore

$$\xi = 0.01\frac{\nu(l+0.6)}{\varphi}$$

where ν = kinematic viscosity of the oil, cm² per sec
l = length of the cushioning channel, cm
φ = coefficient of resistance of the cushioning bore, depending on its cross-sectional shape ($\varphi = 0.029$ cm³ per sec for annular openings).

According to the equation of continuity, the rate of flow during the cushioning period should be

$$v_l f = F v_x$$

where v_x = instantaneous (current) velocity of the piston
f = area of the aperture in the cushioning device.

Hence

$$v_l = v_x \frac{F}{f} \tag{142}$$

Substituting the value of v_l into equation (141) we can write

$$\gamma \frac{F^3}{2gf^2}(1+\xi) v_x^2 = M_0 \frac{d^2x}{dt^2} - \sum S \tag{143}$$

Denoting the product of the constants by

$$R = \gamma \frac{F^3}{2gf^2}(1+\xi)$$

and, substituting $dt = \frac{dx}{v_x}$ and then dividing both sides of the equation by M_0, we obtain

$$\frac{1}{2}\frac{d(v_x)^2}{dx} - \frac{\sum S}{M_0} = \frac{R}{M_0} v_x^2$$

Let $\sum \frac{S}{M_0} = K$ and $\frac{R}{M_0} = K_1$. Then, substituting $v_x^2 = z$, and following general rules, we can write

$$\frac{dz}{z + \frac{K}{K_1}} = 2K_1\, dx \tag{144}$$

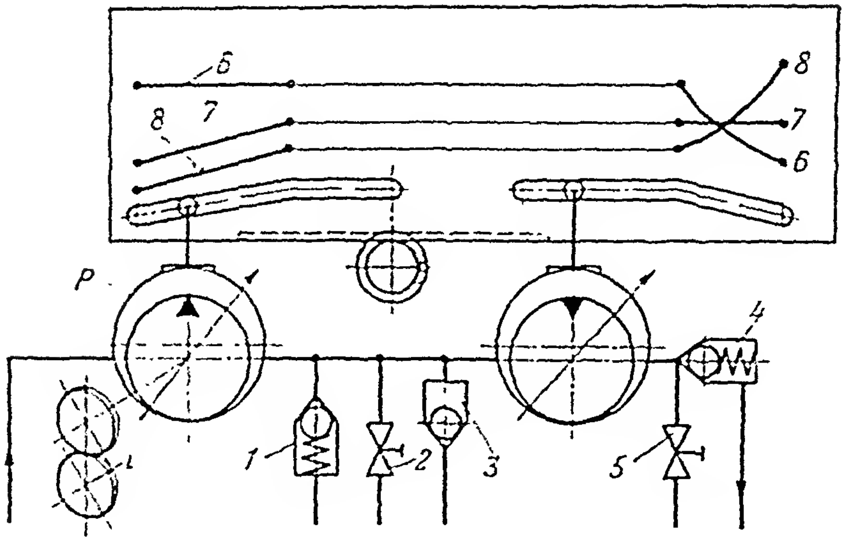


Fig. 105. Diagram of a rotary drive:
P—pump; 1—relief valve; 2—shut-off valve; 3—check valve; 4—backpressure valve; 5—cushioning valve
Characteristic curves: 6—torque; 7—power; 8—speed of hydraulic motor shaft

After integrating and determining the constants on the basis of the initial conditions, where $x = 0$ and $v_x = v_1$, we obtain the equation for the reduction in piston velocity during the period that the work table of the machine is being braked:

$$v_x = \sqrt{\left(v_1^2 - \frac{K}{K_1}\right) e^{2K_1x} - \frac{K}{K_1}}$$

However, since

$$\frac{K}{K_1} = \frac{\sum S}{R}$$

the final form of the equation is

$$v_x = \sqrt{\left(v_1^2 - \frac{\sum S}{R}\right) e^{2K_1x} - \frac{\sum S}{R}} \qquad (145)$$

11-5. Statics and Dynamics of a Rotary Hydraulic Drive

Variable displacement pumps are inversible and can therefore be employed either as consumers of energy, or pumps; or as transmitters of energy, or rotary hydraulic motors.

As a rule in rotary hydraulic drives the hydraulic motor is similar in construction to a pump, though this is not necessarily so. The only positive-displacement pumps that can be used as rotary motors are those in which tangential forces and arms perpendicular to them are obtained when the acting forces are resolved.

The spring of relief valve *1* (Fig. 105) limits the torque of the rotary hydraulic motor. Check valve *3* is opened when the rate of flow of the oil circulating in the system is changed as a result of a change in the pump or hydraulic motor control factor. After one of the control factors has changed, the hydraulic motor continues for a certain time to run by inertia at the same speed but with a changed rate of flow. Suction of air into the system is avoided by valve *3*. When this valve opens, the required amount of oil is admitted into the system so as to compensate for the insufficiency of the circulating volume of oil.

Valve *2* can be used to stop the shaft of the hydraulic motor rapidly without stopping the pump. Together with backpressure valve *4*, cushioning, or damping valve *5* protects the system against shock loads in the periods of bracing and reversal.

The power consumption of the pump is

$$N_1 = N_{em}\eta_1 = CM_{t1}n_1 \text{ kW}$$

where

η_1 = efficiency of the line from the electric motor to the pump
N_{em} = power rating of the pump drive motor
C = factor for converting from kgf-m per sec to kW
N_1 = consumed power of the pump
n_1 = speed of the pump, rpm
M_{t1} = torque of the pump shaft, kgf-m.

Power N_1 is transmitted to the hydraulic motor less the losses. Hence

$$N_2 = N_1\eta_2 = CM_{t2}n_2$$

and

$$M_{t2} = \frac{N_1}{Cn_2}\eta_2 = \frac{N_{em}}{Cn_2}\eta_o \qquad (146)$$

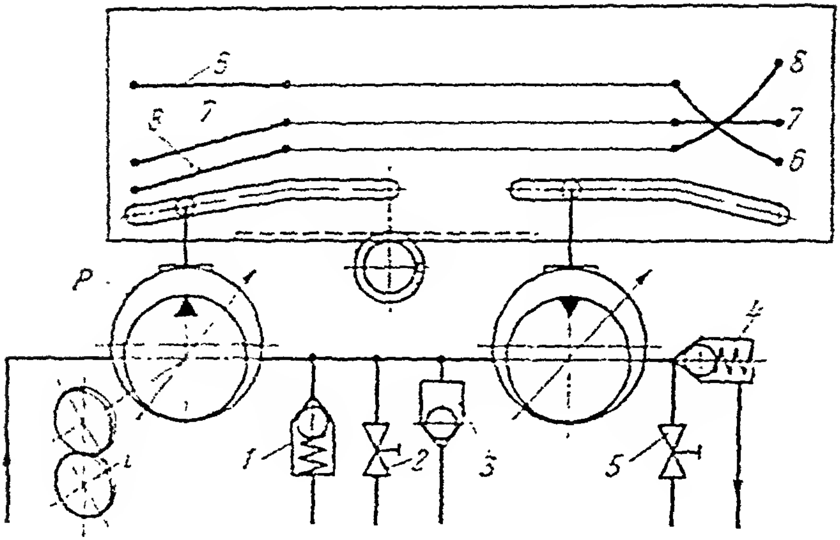
where η_o = $\eta_1\eta_2$ = overall efficiency of the drive
η_2 = efficiency of the hydraulic drive
M_{t2} = torque of the motor

It follows that

$$\frac{M_{t2}}{M_{t1}} = \frac{n_1}{n_2}\frac{\eta}{\eta_1}$$

or, if $n_2\eta_1$ is denoted by n_2^* then

$$\frac{M_{t2}}{M_{t1}} = \frac{n_1}{n_2^*}\eta_o = \frac{K_m}{K_1}\eta_o \qquad (147)$$



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P—[illegible]; 1—relief valve; 2—[illegible] valve; 3—check valve; 4—back pressure valve; 5—cushioning valve; 6—hydraulic [illegible]; 7—[illegible]; 8—speed of hydraulic motor shaft

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$$v_2 = \sqrt{\left(v_1^2 - \frac{K}{K_1}\right) e^{-2K_1 x} - \frac{K}{K_1}}$$

However, since

$$\frac{K}{K_1} = \frac{\sum S}{R}$$

the final form of the equation is

$$v_2 = \sqrt{\left(v_1^2 - \frac{\sum S}{R}\right) e^{-k_1 x} - \frac{\sum S}{R}} \qquad (145)$$

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Variable displacement pumps are invertible and can therefore be employed either as consumers of energy, or pumps or as transmitters of energy, or rotary hydraulic motors.

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where η_o = $\eta_1\eta_2$ = overall efficiency of the drive
η_2 = efficiency of the hydraulic drive
M_{t2} = torque of the motor

It follows that

$$\frac{M_{t2}}{M_{t1}} = \frac{n_1}{n_2}\frac{\eta_o}{\eta_1}$$

or, if $n_2\eta_1$ is denoted by n_2^*, then

$$\frac{M_{t2}}{M_{t1}} = \frac{n_1}{n_2^*}\eta_o = \frac{K_m}{K_t}\eta_o = R$$

where R = conversion factor of the hydraulic drive
K_p and K_m = respective outputs of the pump and hydraulic motor, cu cm per revolution.

These outputs are:

$$K = \frac{\pi}{4} d^2 z 2r \sin\delta = \frac{\pi}{2} d^2 z r \sin\delta$$

or rotary axial piston pumps and hydraulic motors (see Fig. 93), and

$$K = \frac{\pi}{2} d^2 z e$$

for rotary radial piston pumps and hydraulic motors
where d = piston diameter
z = number of pistons in the cylinder barrel of the pump or hydraulic motor
r = piston circle radius
δ = tilt angle of the socket ring or cylinder barrel (see p. 253)
e = eccentricity of the pump.

The values of K_p and K_m characterize the structural dimensions of the operative elements of the pump and hydraulic motor.

If $R = 1$, then $K_p = K_m\eta_o$. This means that if $\eta_o = 1$ the operative elements of the pump and hydraulic motor have identical dimensions, and the speed of the pump would be equal to the maximum speed of the hydraulic motor. The torque developed on the pump shaft would be equal to the torque of the motor. Drives with such characteristics would not convert the torque, and would serve only as fluid couplings.

If $R > 1$, then the working dimensions of the hydraulic motor should be larger than those of the pump, while the speed of the pump should be higher than that of the hydraulic motor. The opposite is true if $R < 1$. A circuit of the latter type is applied in machine tool design for hydraulic headstock spindle drives, for instance in a grinder.

The torque developed by the hydraulic motor or applied to the pump shaft is determined on the basis of the oil flow rate and pressure. Thus

$$M_t = p\frac{Q}{\omega}$$

Since the angular velocity $\omega = 2\pi n$, and the delivery (if there are no volumetric losses) $Q = Kn$, where K is the output per revolution, the preceding equation can be written as

$$M_t = p\frac{K}{2\pi} \tag{147}$$

where p is the pressure drop in the pump (hydraulic motor).

We shall again resort to the d Alembert equation to determine certain dynamic characteristics of rotary hydraulic drives Thus

$$J_t \frac{d\omega}{dt} = M_t \tag{148}$$

where ω = angular velocity of the hydraulic motor shaft
J_t = moment of inertia of the rotating masses referred to the shaft of the hydraulic motor
M_t = referred torque of the hydraulic motor

$$J_t = J_{cb} + J_1 \left(\frac{\omega_1}{\omega}\right)^2 + J_2 \left(\frac{\omega_2}{\omega}\right)^2 - \tag{149}$$

where J_{cb} — moment of inertia of the cylinder barrel assembly of the hydraulic motor i e the shaft to which moment of inertia is referred
J_1 = moment of inertia of the first shaft
ω_1 = angular velocity of the first shaft
J_2 = moment of inertia of the second shaft
ω_2 = angular velocity of the second shaft

If J_1 = const then equation (148) can be written as

$$\frac{d}{dt} = \frac{M_t}{J_t} = \text{const} \tag{150}$$

which indicates that motion proceeds with constant acceleration and the angular velocity is

$$\omega = \frac{M_t}{J_t} t \tag{151}$$

It follows that if ω_s is the steady angular velocity of the hydraulic motor the time required for it to accelerate to this velocity is

$$T_0 = \frac{J_t}{M_t} \omega_s \tag{152}$$

The time T_0 characterizes the inertia of the drive and is of prime importance in appraising its dynamic qualities

The speed of response of a drive is characterized by the angle of acceleration Θ It can be determined from equation (151) Thus

$$\frac{d\varphi}{dt} = \frac{M_t}{J_t} t \tag{153}$$

$$\Theta = \int_0^{T_0} \frac{M_t}{J_t}\, dt = \frac{M_t}{2J_t} T_0^2$$

or finally

$$\Theta = \frac{1}{2} \frac{J_t}{M_t} \omega_s^2 \tag{154}$$

In most cases, the time required to accelerate the cylinder barrel of a pump or hydraulic motor to running speed is expressed by the number of revolutions it makes during this time; the less the number of revolutions, the quicker the barrel reaches the steady speed.

The number of revolutions during the acceleration time is

$$n_{0x} = \int_0^{T_0} \frac{n_t}{60}\, dt \tag{155}$$

where $\frac{n_t}{60}$ is the current speed, revolutions per sec, of the cylinder barrel during the acceleration period.

Substituting $\frac{d\omega}{dt} = \frac{\pi dn_t}{30 dt}$ in equation (150), we have

$$dt = J_t \frac{\pi}{30} \frac{dn_t}{M_t}$$

and then equation (155) can be written as

$$n_{0x} = J_t \frac{\pi}{30 \cdot 60} \int_0^{n} \frac{n_t}{M_t}\, dn_t \tag{156}$$

where M_t is the referred torque as before

CHAPTER 12

SPEED CONTROL OF HYDRAULIC PISTONS AND ROTARY HYDRAULIC MOTORS

12-1. General Principles

The purpose of either manual or automatic controls is to obtain and maintain the specified operating cycle of the machine (or machine tool in our case) Thus, control can be defined as the process of establishing the given parameters—pressure, velocity, rpm, etc —and maintaining them at the required levels at each moment of the cycle

The operating conditions of a machine tool depend on many interrelated parameters, and may vary continuously or periodically. To maintain more or less constant operating conditions, i e ones that vary only within specified limits or according to a given law, it is necessary to operate the controls, either manually or automatically

Manual control is usually confined to a single parameter which is the most effective in varying the operating conditions of the machine tool

Automatic control may be accomplished either in respect to a single parameter or to several parameters

The mechanism that is to be controlled is called the *controlled member*, while the device directly accomplishing the controlling process is called the *controller* Taken together the controlled member and controller are called the *control system*

The level and quality of the controlled process are related to a single or several parameters such as pressure, velocity, acceleration, travel, etc. These are called the *controlled parameters*

There are two general types of automatic control In one, control is based upon the parameters of the controlled process itself use being made of various protective apparatus hydraulic speed stabilizers etc In the other case, the process is controlled by parameters that are introduced from outside This is the so called *programmed control*

The speed of a hydraulic motor can be controlled by changing the amount of fluid flowing through it in unit time this being accomplished by

(a) changing the operating conditions of the pump by varying its displacement,

(b) changing the resistance but at constant pressure of a part of the line through which the fluid flows

Both of these methods are known as *volume controls*; the first is called *variable-displacement* controls and the second—*flow* controls. This can be effected in either case by:

(1) varying the flow of fluid in the line transmitting energy to the hydraulic motor, called metering-in controls;

(2) varying the rate of discharge from the hydraulic motor, called metering-out controls.

Because of certain difficulties and a higher cost, as compared to other control methods, a combination of variable-displacement and metering-out controls is rarely applied, and is not a typical method in machine tool engineering.

The choice of the control method should be based on the power, required pressure, nature of variation of the useful load, type of pump to be installed, pump characteristics and certain other factors.

The principal feature of variable-displacement control is that, at a constant load, the power output of the hydraulic motor is proportional to the delivery of the pump. A rotary hydraulic motor is to be loaded, in this case, by a constant torque, and a hydraulic piston—by a constant pulling or pushing force. This control method is extensively employed in various branches of machine building, and especially for machines which require a large pulling or pushing force, or maximum torque in starting.

In metering-in or metering-out flow controls, the pressure and delivery of the pump are constant as, consequently, is its power consumption. The speed of the hydraulic motor depends upon the resistance of the flow control device. The excess part of the oil drains continuously back to the tank through the relief valve without performing any useful work.

When the speed of the hydraulic motor is reduced, the part of the pump delivery draining back to the tank increases, raising the losses. When the motor speed is increased, losses are reduced. Thus, flow control is based on the variation of the magnitude of the losses, i.e. on the variation of the efficiency of the hydraulic system. Therefore, generally speaking, flow control is justified only at low power rating of the hydraulic motor. On the other hand, due to the surplus delivery of the pump, volumetric losses affect the kinematic rigidity of the hydraulic system to a lesser extent than in variable-displacement control.

The *supply* of a rotary hydraulic motor or cylinder may be either *individual* or *centralized*. Any method of control (variable-displacement or flow control) is applicable to individual supply. Only flow control can be employed with a centralized supply since the oil flow to each power member should be distributed in accordance with the resistances of the pressure lines.

Features of individual supply (Fig. 106, and also see Fig. 183, Vol. 1) where pump P_2 is for handling operations and pump P_{td} is for the table drive) of several hydraulic motors are the independent pressures, p_1, p_1'

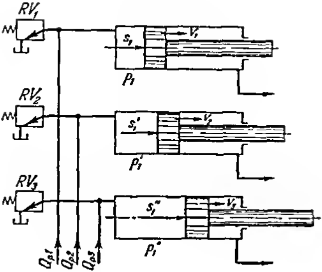

Fig 106 Individual supply of a hydraulic circuit: RV_1, RV_2 and RV_3—relief valves

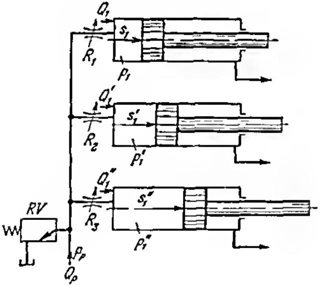

Fig 107 Centralized supply of a hydraulic circuit: RV—relief valve, R_1, R_2 and R_3 variable hydraulic restrictions

and p_1'' in the pressure lines of each motor; independent velocities v_1, v_1' and v_1'' and travel; and the independent power consumption. A supply system of this type, however, requires separate and independent power units (pumping stations) and independent control apparatus. This substantially raises the cost of the machine tool and increases its overall size. For these reasons, systems of individual supply are rarely applied in machine tool engineering.

A centralized system of supply has a single pump accommodating several hydraulic motors (Fig. 107). There is only one pressure in the system—the maximum pressure p_p—which is determined by the required pulling (or pushing) force S and piston area F. If the hydraulic motors operate separately, the pump delivery Q_p is determined by the motor with the highest speed. If the hydraulic motors operate simultaneously, the pump delivery should equal the arithmetic sum of the required displacements of all the simultaneously operating motors: $Q_p = Q_1 + Q_1' + Q_1''$. Variable restrictions R_1, R_2 and R_3 are provided in the lines for a tentative distribution of the pump delivery in accordance with the *speeds and loads* of the hydraulic motors. Any change, however, in the useful load of any hydraulic motor will lead to a redistribution of the pump delivery among the motors.

If the loads vary, stable travel can be achieved by the use of speed stabilizers (Fig. 120) which are usually pressure-compensated flow-control valves. Their simplicity and small size have won such supply systems widespread application in hydraulic machine tools.

To calculate the areas of the apertures in restrictors, it proves convenient to introduce the conception of conductivity s of the restrictor. Then equation (84) can be written as

$$Q = s\sqrt{\Delta p} \tag{157}$$

where s is the conductivity of the restrictor.

$$s = \frac{1}{\sqrt{R}} = f\mu\sqrt{2\frac{g}{\gamma}}$$

If the restrictors are connected in parallel (Fig. 107), the pressure drops will be equal in all branches, and the delivery for the whole system is

$$Q_t = s_0\sqrt{\Delta p}$$

where s_0 is the total conductivity of the restrictors

$$s_0 = \sum_{i=1}^{n} s_i$$

For a system such as shown in Fig. 107, $i = 1$, 2 and 3.

If it is assumed that the flow coefficient μ is the same for all restrictors connected in parallel, then

$$s_0 = \sqrt{\frac{2g}{\gamma}}\,\mu(f_1 + f_2 + f_3)$$

where f_1, f_2 and f_3 are the areas of the aperture in the restrictors

12-2. Variable-Displacement Speed Control of Hydraulic Motors

In considering this method of speed control, as well as other methods further on, it is assumed that the oil contains no air and that there is a linear relationship between the external and internal volumetric losses in the system (in the pump, hydraulic motor and control devices) and the pressure

The pressure p_1 in the cylinder (Fig 108) is determined by equation (121) Provision is usually made for a backpressure of $p_2 \simeq 1.5$ kgf per sq cm, produced by means of valve *3*, to stabilize the friction forces and, therefore, value p_s in equation (121) The spring of relief valve *2* is adjusted to the maximum pressure p_0 of pump *1* Directional control valve *5* changes the direction of piston travel It is advisable to install check valve *4* in the pressure line to prevent the system from emptying completely, and air from entering when the pump is switched off

Figure 109 shows the characteristic curves for this control method It can be seen that the power is proportional to the load applied to the piston Since volumetric losses are inevitable, any increase in pressure results in a reduction in speed

Since there are no power losses from throttling in variable-displacement control, the oil is heated less This increases the efficiency of the system This speed control method proves expedient for comparatively high power ratings, and when a wide range of speed variation is required

A certain amount of power is required to start the pump controls Depending upon the size of the drive and the output power of the hydraulic motor, this starting power is $N_s \simeq (0.02$ to $0.1)\,N$, where N is the output power of the hydraulic motor

Making use of the continuity equation we can write for a *reciprocating* drive (Fig 108)

$$Q_x = K_p n\psi = v_1 F_1 + q_\sigma \qquad (158)$$

where K_p = specific displacement of the pump, cu cm per revolution

ψ = pump control factor, equal to the ratio of the current value of the controlled pump parameter to the maximum value of this parameter (thus for example, for radial piston pumps and

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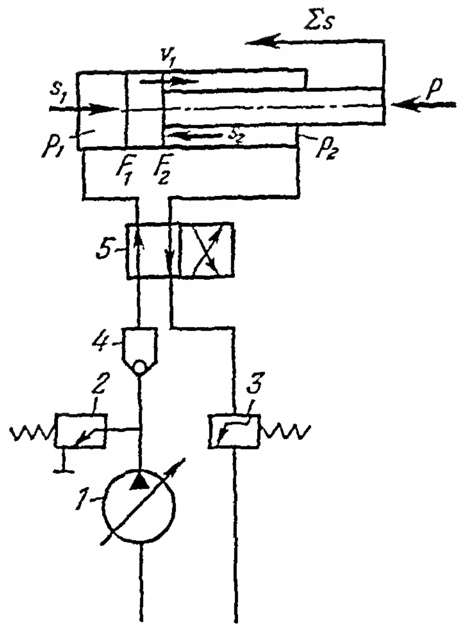

Fig. 108. Variable-displacement piston speed control:
1—variable-displacement pump; 2—relief valve; 3—backpressure valve; 4—check valve; 5—two-position directional valve

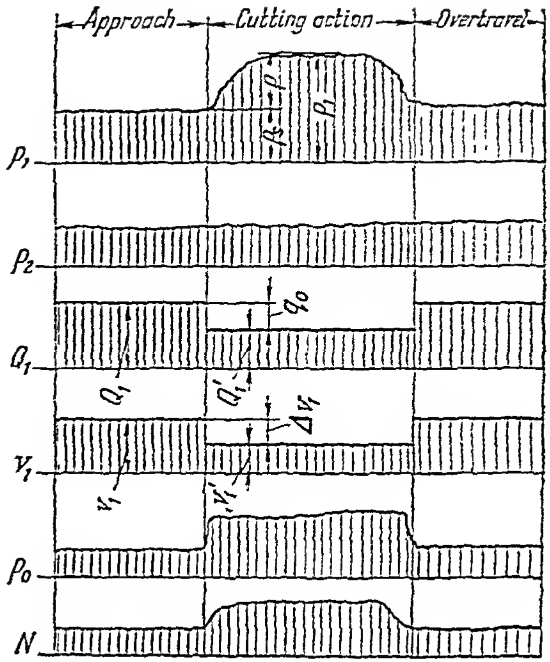

Fig. 109. Pressure variation and piston speed control with a variable-displacement pump

hydraulic motors $\psi = \frac{e_x}{e}$, where e_x is the current eccentricity, and for axial piston pumps and hydraulic motors $\psi = \frac{\sin \delta_x}{\sin \delta}$, where δ_x is the current angle of tilt of the cam plate, cylinder barrel, etc., in accordance with the construction of the pump and its mechanism for controlling the displacement)

F_1 = piston area

q_σ = volumetric losses in the hydraulic system determined by equation (86).

The preceding equation is used to find the actual velocity of the piston at the pressure p_1. Thus

$$v_1 = \frac{K_p}{F_1} n\psi - \frac{q_\sigma}{F_1} \tag{159}$$

If $v_0 = \frac{K_p}{F_1} n\psi$ is the geometrical velocity of the piston, i.e. the velocity when the pressure $p = 0$, then the difference in velocities $(v_0 - v_1)$ is the

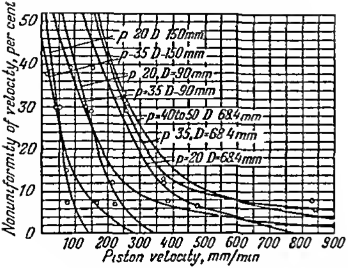

Fig 110 Nonuniformity of piston travel in variable-displacement speed control

duction in piston velocity due to volumetric losses

$$\Delta v_1 = v_0 - v_1 = \frac{q_0}{F_1} \tag{160}$$

Volumetric losses reduce the piston velocity, but their effect is decreased some extent by a corresponding increase in the piston area F_1 This pro-dure is applied in machine tool engineering

The quality of the hydraulic system of a machine tool is appraised by the gree of nonuniformity χ of piston velocity in a reciprocating drive or of otor rotation speed in a rotary drive

$$\chi = \frac{\Delta v_1}{v_0} = 1 - \frac{v_1}{v_0} = \frac{q_0}{v_0 F_1} \tag{161}$$

The curves of Fig 110 show how the degree of nonuniformity of piston locity varies with pressure p for various piston diameters D and velocities is evident from the curves that the piston diameter greatly affects the iformity of travel For instance, at a velocity of $v_0 = 100$ mm per min d a pressure of $p = 20$ kg per sq cm, the degree of nonuniformity is $\chi = 6$ r cent for a piston diameter of $D = 150$ mm, it increases to $\chi = 38$ per nt for $D = 90$ mm, and at $D = 68$ mm the nonuniformity $\chi > 50$ per nt, a value that is absolutely unacceptable for machine tools

The reduction in velocity Δv_1 of the hydraulic motor will be

$$\Delta v_1 = \sigma \frac{p}{F_1} \tag{162}$$

where p is the pressure in the system.
Hence the load may be obtained as

$$P = \frac{\Delta v_1}{\sigma} F_1^2 \tag{163}$$

Consequently, the kinematic rigidity of the system (see p. 224) is

$$\frac{\partial P}{\partial v_1} = -\frac{F_1^2}{\sigma_p} = -j_k \tag{164}$$

Hydraulic machine tools, operating with a high pulling (or pushing) force and with a periodically varying load, are appraised on the basis of the dynamic rigidity of the system. Its magnitude indicates whether the piston area F_1, cylinder length, length of the piping, its elasticity, etc., have been rationally chosen. We can determine piston travel under the action of a load by setting up the equation for volumetric equilibrium of the power cylinder (see Fig. 108):

$$F_1 x - F_1 v_0 t + 2(\Theta_0 + cF_1 l_1)\frac{P}{F_1} = 0 \tag{165}$$

Hence, the permissible useful load is

$$P = \frac{F_1^2 (v_0 t - x)}{2(\Theta_0 + cF_1 l_1)} \tag{166}$$

and the dynamic rigidity of the system is

$$j_d = \frac{\partial P}{\partial x} = -\frac{F_1^2}{2(\Theta_0 + cF_1 l_1)} \tag{167}$$

Thus, the larger the elastic constant Θ_0 of the system, the less, all other conditions being equal, the dynamic rigidity j_d will be and the more reasons for the development of vibration upon changes in the load P.

In deriving the rigidity equation, it was assumed that the volumetric losses are negligible quantities. Practically, they are inevitable in the system, and in the given case they promote a certain amount of damping of the vibration. The rigidity equations (164 and 167) show that the length of the power cylinder and its cross-sectional area are of prime importance in questions concerning the stability.

The dynamic rigidity of a hydraulic system must be known for a tentative determination of the natural frequency of vibration in the system. Thus

$$n_n = \frac{1}{2\pi}\sqrt{\frac{j_d}{M}} \tag{168}$$

where M is the referred mass of the moving parts.

Data on the natural frequency of the system and its level are required in dynamic investigations of the system, for selecting valve springs, designing dampers, and also for determining the possible frequency pass band.

Applying the continuity equation to a rotary drive (Fig. 105) as well, we obtain

$$K_p \eta_v n_{em} \psi_p = K_m i_1 n_2 \psi_m \tag{169}$$

from which the speed of the rotary hydraulic motor is

$$n_2 = \frac{K_p}{K_m} n_{em} \frac{1}{i_1} \eta_v \frac{\psi_p}{\psi_m} = \frac{\eta_v}{R} n_{em} \frac{1}{i_1} \frac{\psi_p}{\psi_m} \tag{170}$$

where ψ_p and ψ_m = pump and hydraulic motor control factors, respectively, as in equation (158)

$i_1 = \frac{n_{em}}{n_1}$ = gearing ratio between the electric motor and pump

n_{em} = speed of the electric motor

n_1 = speed of the pump rotor

η_v = volumetric efficiency of the drive

$R = \frac{K_m}{K_p}$ = conversion factor of the drive

It is assumed for tentative calculations that the volumetric efficiency η_v is constant and equal to the mean value over the whole range of speed control.

Denoting the product of the constants by c,

$$\eta_v \frac{n_{em}}{i_1} = c \tag{171}$$

we obtain the equation for the minimum speed of the rotary hydraulic motor shaft

$$n_{2min} = c \frac{1}{R} \frac{\psi_{p\,min}}{\psi_{m\,max}} \tag{172}$$

The transition or critical, speed, i.e. transition from the range of pump control to the range of rotary motor control is determined at the maximum ratio of the control factors this being equal to unity. Then

$$n_{2cr} = c \frac{1}{R} \tag{173}$$

The maximum speed of the rotary hydraulic motor is

$$n_{2max} = c \frac{1}{R} \frac{\psi_{p\,max}}{\psi_{m\,min}} \tag{174}$$

In hydraulic couplings (see p. 272) $R = \frac{K_m}{K_p} = 1$ and $i_1 = 1$, then

$$K_m = K_p, \quad n_{2cr} = \eta n_{em} \text{ and } n_1 = n_{2max}$$

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The reduction in velocity Δv_1 of the hydraulic motor will be

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where p is the pressure in the system.
Hence the load may be obtained as

$$P = \frac{\Delta v_1}{\sigma} F_1^2 \tag{163}$$

Consequently, the kinematic rigidity of the system (see p. 224) is

$$\frac{\partial P}{\partial v_1} = -\frac{F_1^2}{\sigma_p} = -j_k \tag{164}$$

Hydraulic machine tools, operating with a high pulling (or pushing) force and with a periodically varying load, are appraised on the basis of the dynamic rigidity of the system. Its magnitude indicates whether the piston area F_1, cylinder length, length of the piping, its elasticity, etc., have been rationally chosen. We can determine piston travel under the action of a load by setting up the equation for volumetric equilibrium of the power cylinder (see Fig. 108):

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In deriving the rigidity equation, it was assumed that the volumetric losses are negligible quantities. Practically, they are inevitable in the system, and in the given case they promote a certain amount of damping of the vibration. The rigidity equations (164 and 167) show that the length of the power cylinder and its cross-sectional area are of prime importance in questions concerning the stability.

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from which the speed of the rotary hydraulic motor is

$$n_2 = \frac{k_p}{k_m} n_{em} \frac{1}{i_1} \eta_v \frac{\psi_p}{\psi_m} = \frac{\eta_v}{R} n_{em} \frac{1}{i_1} \frac{\psi_p}{\psi_m} \tag{170}$$

where ψ_p and ψ_m = pump and hydraulic motor control factors, respectively, as in equation (158)

$i_1 = \frac{n_{em}}{n_1}$ = gearing ratio between the electric motor and pump

n_{em} = speed of the electric motor

n_1 = speed of the pump rotor

η_v = volumetric efficiency of the drive

$R = \frac{k_m}{k_p}$ = conversion factor of the drive

It is assumed for tentative calculations that the volumetric efficiency η_v is constant and equal to the mean value over the whole range of speed control.

Denoting the product of the constants by c,

$$\eta_v \frac{n_{em}}{i_1} = c \tag{171}$$

we obtain the equation for the minimum speed of the rotary hydraulic motor shaft

$$n_{2min} = c \frac{1}{R} \frac{\psi_{p\,min}}{\psi_{m\,max}} \tag{172}$$

The transition, or critical, speed, i.e. transition from the range of pump control to the range of rotary motor control is determined at the maximum ratio of the control factors, this being equal to unity. Then

$$n_{2cr} = c \frac{1}{R} \tag{173}$$

The maximum speed of the rotary hydraulic motor is

$$n_{2max} = c \frac{1}{R} \frac{\psi_{p\,max}}{\psi_{m\,min}} \tag{174}$$

In hydraulic couplings (see p. 272) $R = \frac{k_m}{k_p} = 1$ and $i_1 = 1$, then

$$k_m = k_p, \quad n_{2cr} = \eta n_{em} \text{ and } n_1 = n_{2max}$$

Data on the natural frequency of the system and its level are required in dynamic investigations of the system, for selecting valve springs, designing dampers, and also for determining the possible frequency pass band

Applying the continuity equation to a rotary drive (Fig 105) as well, we obtain

$$K_p \eta_v n_{em} \psi_p = K_m i_1 n_2 \psi_m \tag{169}$$

from which the speed of the rotary hydraulic motor is

$$n_2 = \frac{K_p}{K_m} n_{em} \frac{1}{i_1} \eta_v \frac{\psi_p}{\psi_m} = \frac{\eta_v}{R} n_{em} \frac{1}{i_1} \frac{\psi_p}{\psi_m} \tag{170}$$

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The reduction in velocity Δv_1 of the hydraulic motor will be

$$\Delta v_1 = \sigma \frac{p}{F_1} \tag{162}$$

where p is the pressure in the system.
Hence the load may be obtained as

$$P = \frac{\Delta v_1}{\sigma} F_1^2 \tag{163}$$

Consequently, the kinematic rigidity of the system (see p. 224) is

$$\frac{\partial P}{\partial v_1} = -\frac{F_1^2}{\sigma_p} = -j_k \tag{164}$$

Hydraulic machine tools, operating with a high pulling (or pushing) force and with a periodically varying load, are appraised on the basis of the dynamic rigidity of the system. Its magnitude indicates whether the piston area F_1, cylinder length, length of the piping, its elasticity, etc., have been rationally chosen. We can determine piston travel under the action of a load by setting up the equation for volumetric equilibrium of the power cylinder (see Fig. 108):

$$F_1 x - F_1 v_0 t + 2(\Theta_0 + cF_1 l_1)\frac{P}{F_1} = 0 \tag{165}$$

Hence, the permissible useful load is

$$P = \frac{F_1^2 (v_0 t - x)}{2(\Theta_0 + cF_1 l_1)} \tag{166}$$

and the dynamic rigidity of the system is

$$j_d = \frac{\partial P}{\partial x} = -\frac{F_1^2}{2(\Theta_0 + cF_1 l_1)} \tag{167}$$

Thus, the larger the elastic constant Θ_0 of the system, the less, all other conditions being equal, the dynamic rigidity j_d will be and the more reasons for the development of vibration upon changes in the load P.

In deriving the rigidity equation, it was assumed that the volumetric losses are negligible quantities. Practically, they are inevitable in the system, and in the given case they promote a certain amount of damping of the vibration. The rigidity equations (164 and 167) show that the length of the power cylinder and its cross-sectional area are of prime importance in questions concerning the stability.

The dynamic rigidity of a hydraulic system must be known for a tentative determination of the natural frequency of vibration in the system. Thus

$$n_n = \frac{1}{2\pi}\sqrt{\frac{j_d}{M}} \tag{168}$$

where M is the referred mass of the moving parts.

Data on the natural frequency of the system and its level are required in dynamic investigations of the system, for selecting valve springs, designing dampers, and also for determining the possible frequency pass band.

Applying the continuity equation to a rotary drive (Fig. 105) as well, we obtain

$$K_p \eta_v n_{em} \psi_p = K_m i_1 n_2 \psi_m \tag{169}$$

from which the speed of the rotary hydraulic motor is

$$n_2 = \frac{K_p}{K_m} n_{em} \frac{1}{i_1} \eta_v \frac{\psi_p}{\psi_m} = \frac{\eta_v}{R} n_{em} \frac{1}{i_1} \frac{\psi_p}{\psi_m} \tag{170}$$

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$R = \frac{K_m}{K_p}$ = conversion factor of the drive.

It is assumed for tentative calculations that the volumetric efficiency η_v is constant and equal to the mean value over the whole range of speed control.

Denoting the product of the constants by c,

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$$n_{2min} = c \frac{1}{R} \frac{\psi_{p\,min}}{\psi_{m\,max}} \tag{172}$$

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In hydraulic couplings (see p. 272), $R = \frac{K_m}{K_p} = 1$ and $i_1 = 1$, then

$$K_m = K_p, \quad n_{2cr} = \eta n_{em} \text{ and } n_1 = n_{2max}$$

If the torque can be converted, then $R > 1$, i.e. $K_m > K_p$ and

$$n_1 > n_{2max}$$

For the same electric motor speed a gearing-up arrangement ($i_1 > 1$) is required and therefore the critical speed should be

$$n_{2cr} = \eta \frac{n_{em}}{Ri_1} \tag{175}$$

In high-speed drives $R < 1$, i.e. $K_m < K_p$, and

$$n_1 < n_{2max}$$

In this case, with the same electric motor speed, a gearing-down arrangement in which $i_1 < 1$ is required.

The maximum speed of a rotary hydraulic motor depends upon the construction of the drive and the quality of its manufacture. The maximum speed of radial-piston drives does not usually exceed 1,600 rpm. Axial-piston hydraulic motors run at higher speeds, up to 2,500 rpm, because the mass of the cylinder barrel assembly of such pumps and motors is symmetrically distributed in reference to the axis of rotation.

The quality of manufacture of a drive is characterized by the minimum speed of the rotary hydraulic motor at full load. It may be as low as $n_{2min} = 40$ to 50 rpm at a pressure of 60 to 65 kgf per sq cm.

The static characteristic curves of a rotary drive are shown in Fig. 111. The speed control range of the rotary hydraulic motor does not exceed 1 : 3, as a rule, while the control range of the pump is considerably higher—from 1 : 400 to 1 : 450. Variable-displacement pump control (Fig. 111*a*) maintains a constant torque M_{tm} on the shaft of the hydraulic motor, and a linear variation of power consumption N and speed n_2 proportional to the rate of flow. Variable-displacement motor control (Fig. 111*b*) maintains a constant power consumption with a varying torque on the hydraulic motor shaft. A characteristic curve of this type is desirable in machine tools having a hydraulic drive for spindle rotation.

The volumetric efficiency of a rotary hydraulic drive can be readily determined from the idling speed n_{2i} and speed n_{2l} under load of the hydraulic motor shaft. Thus

$$\eta_v = \frac{n_{2i} - n_{2l}}{n_{2i}} = 1 - \frac{n_{2l}}{n_{2i}}$$

A hydraulic circuit with closed oil circulation (Fig. 112, see also Figs. 201, 207 and 209 of Vol. 1) is employed in cases when it is necessary to substantially reduce the capacity of the oil tank in small-size high-speed machine tools or in control mechanisms (especially in transportation machinery), as well as to stabilize the friction forces to some extent. This circuit re-

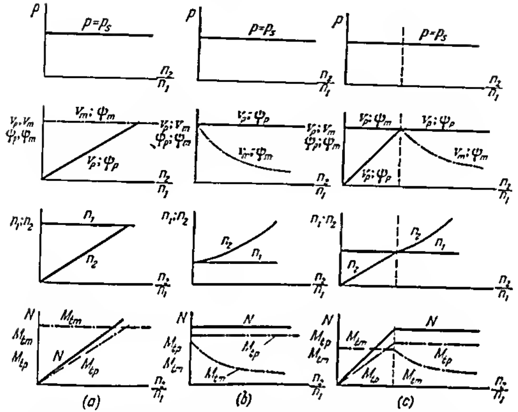


Fig 111 Static characteristic curves of a rotary drive
(a) variable-displacement pump and fixed-displacement rotary hydraulic motor (b) fixed-displacement pump and variable-displacement rotary hydraulic motor (c) variable-displacement pump and hydraulic motor (combination controls)

quires two pumps low-pressure pump *8* and high-pressure pump *1*. The latter is usually of the variable-displacement type The low-pressure pump supplies the high-pressure pump and simultaneously compensates for volumetric losses in the system, as well as for the differences in volumes on the right and left sides of the piston (if a single-end rod cylinder is used). Moreover, this pump develops a backpressure which levels out the variable friction force to some degree

The high-pressure pump provides the required pulling force and controls the piston speed

If a single-end rod piston is employed, as in Fig 112, which travels at a velocity v_1, then the circulating rate of flow in the system is

$$Q_{r1} = v_1 F_1$$

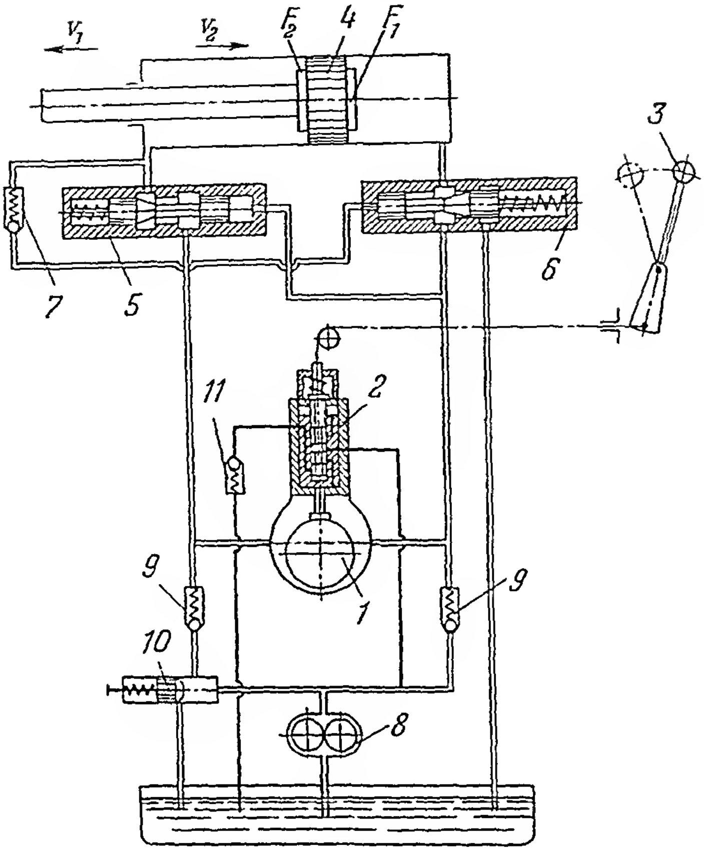


Fig. 112. Speed control in a hydraulic circuit with closed oil circulation:
1—variable-displacement pump; 2—pump displacement control gear; 3—control lever; 4—power piston; 5 and 6—chokes for braking the power piston; 7—check valve; 8—fixed-displacement auxiliary pump; 9—check valve; 10—relief valve; 11—backpressure valve

Since the piston areas are not equal on the two sides, the differences in displacements and all the volumetric losses q_σ in the system are compensated by the delivery of pump 8. Hence

$$Q_{p2} = v_1 (F_1 - F_2) + q_\sigma \tag{176}$$

As the piston travels in the opposite direction, a surplus of circulating flow occurs, since the discharge Q_{p2} from the cylinder exceeds the input Q_{p1}, i.e. $Q_{p2} > Q_{p1}$. Part of this surplus oil drains back to the tank while the remainder compensates for volumetric losses. In this case

$$\Delta Q = v_2 (F_1 - F_2) - \sigma p_2 \tag{177}$$

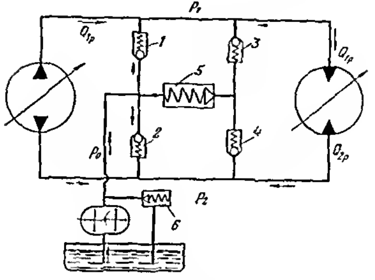


Fig. 113. Speed control of a rotary drive in a hydraulic circuit with closed oil circulation
1, 2, 3 and 4—check valves; 5—high pressure relief valve; 6—relief valve of make-up pump

A system of closed circulation for hydraulic rotary or reciprocating drives requires prefilling. Therefore air bleeding plugs must be provided for releasing air trapped in the system. Air pockets in the system sharply upset uniform operation of the hydraulic motor and in some cases they interrupt oil circulation.

Closed circulation is widely applied in rotary drives (Fig. 113) due to the compact and simple facilities they provide for controlling the displacement of the pump or hydraulic motor.

The shaft of the rotary hydraulic motor is reversed either by shifting the slide block of the pump or motor through the neutral position, or by reversing the flow with a valve. In the latter case the transition process in reversal proceeds considerably faster than in reversal by shifting the slide block.

An interchange in pressures occurs in the lines between the pump and motor in a reversible drive (Fig. 113). This is accomplished by check valves *1* and *2* having the same purpose as valve *3* (Fig. 10?) and capable of replenishing small losses in flow. Low delivery gear pumps are used for rapid replenishment and to maintain a constant make-up pressure. The level of pressure of these pumps and, consequently, the level of the make-up pressure is determined by the make-up relief valve *6*.

Valves *3* and *4* enable the high pressure safety relief valve *5* to operate in a reversible cycle. Upon an overload the rotary hydraulic motor stops

but the pump continues to operate, its full delivery passing through valve *3* or *4*, in accordance with the direction of rotation, and further through safety relief valve *5* and check valve *2* or *1* into the suction chamber of the pump.

A gear pump can be used, moreover, for controlling the servomechanism which varies pump displacement, for example, in high-power drives, with automatic controls, etc.

Safety relief valve *5* (Fig. 113) limits the maximum pressure of the pump. In idling, the discharge pressure p_1 is greater than the backpressure p_2 and the consumed power is expended only in overcoming the friction resistance.

Upon an increase in pressure, for instance, when the load is switched in, the backpressure p_2 drops, check valve *2* admits oil and the make-up pressure p_0 is equalized with the backpressure p_2. At $p_0 = p_2$, valve *2* closes and the delivery of the gear pump is drained back to the tank. This is repeated each time the load changes.

When a load is applied, the rate of flow through the pump (subscript p) is equal to

$$Q_{1p} = K'_p \omega_p - \sigma_p (p_1 - p_2) - \sigma'_p p_1 \text{ cu cm per min} \tag{178}$$

where Q_{1p} = pump delivery to the pressure line, cu cm per min

K'_p = pump displacement, cu cm per rad, equal to $K'_p = \frac{K_p}{2\pi}$ (value of K_p is given on p. 279)

ω_p = angular velocity of the pump shaft, rad per min

σ_p = coefficient of internal volumetric losses of the pump, cm^5 per kgf-min

σ'_p = coefficient of external volumetric losses of the pump, cm^5 per kgf-min (here p_1 and p_2 are given in kgf/cm^2).

Likewise, the rate of flow passing through the hydraulic motor (subscript m) at the same load is

$$Q_{2m} = K'_m \omega_m + \sigma_m (p_1 - p_2) + \sigma'_m p_1 \text{ cu cm per min} \tag{179}$$

Equating the right-hand sides of the last two equations we can write

$$\omega_m = \frac{1}{K'_m} [K'_p \omega_p - \Delta p (\sigma_p + \sigma_m) - p_1 (\sigma'_p + \sigma'_m)] \tag{180}$$

or, substituting the pump control factor ψ_p for the specific displacement K'_p, we obtain

$$\omega_m = \frac{1}{K'_m} [k_b \psi_p \omega_p - \Delta p (\sigma_p + \sigma_m) - p_1 (\sigma'_p + \sigma'_m)] \tag{181}$$

where k_b is the specific displacement of the pump per unit of the pump control factor, i.e. $k_b = \frac{K'_p}{\psi_p}$.

By equating the angular velocity of the hydraulic motor to zero ($\omega_m = 0$) we obtain the minimum control factor for the given pressure drop $\Delta p = p_1 - p_2$ Thus

$$\psi_{p\,min} = \frac{1}{k_b\omega_p}[\Delta p(\sigma_p + \sigma_m) - p_1(\sigma_p + \sigma_m)] \tag{182}$$

Internal volumetric losses are due to the pressure drops and to clearances between contacting surfaces in movable and fixed joints These losses remain in the system but do not participate in the transmission of energy to the hydraulic motor

External losses are losses through sealing facilities that are in direct contact with the atmosphere (packing glands of the piston rod, pump shafts, motor shafts etc)

A fu using an arrangement ents most of the power through a differential mechanism and the lesser part through a hydraulic transmission

The two input powers are summated on the output shaft of the differential mechanism In such drives, the torque and speed of the output shaft are controlled hydraulically by a variable displacement pump

12-3 Flow Control of Hydraulic Motors

A flow control valve is a local hydraulic resistance, specially designed for controlling the speed of a rotating or reciprocating hydraulic motor and installed in the line of fluid flow

The feature that distinguishes a flow control valve from ordinary local restrictions for instance pressure control valves is that its orifice for fluid flow is varied only by external action while the orifice of a pressure control valve is varied by the action of the fluid flowing through

The controlling action of a flow control valve is based on the fact that the hydraulic resistance varies with variation in the orifice This changes the pressure drop and consequently the rate of oil flow through the valve

The resistance coefficient of a flow control valve depends upon many factors, such as the sharpness of the edges of the throttling orifice its shape in the cross and longitudinal sections the oil temperature etc Consequently, the flow control process is quite complex

The hydraulic resistance R_{th} of a flow control valve should be greater than that of the piping and relief valve R_v (see Figs 114 115 and 116) The less sharp the edges of the throttling orifice the fewer its resistance Sharpness of the edges is usually measured in relative values—the ratio of the radius r of the entrance edge to the diameter d of the intake hole This

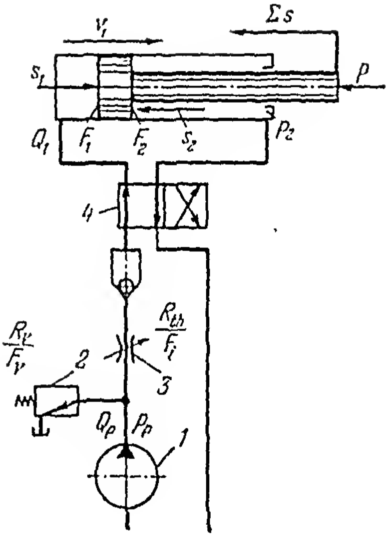


Fig. 114. Metering-in circuit for controlling piston speeds:
1—pump; *2*—relief valve; *3*—flow control valve; *4*—two-position directional valve

ratio varies with variation of the throttling orifice. Thus, the larger the orifice, the less the effect of the sharpness of the edges on its resistance.

It has been shown experimentally that at $\frac{r}{d} > 0.2$, the sharpness of the edges has almost no effect on the resistance of the flow control valve.

Distortions of the fluid stream sharply affect the characteristics of a flow control valve. It is therefore advisable to install a flow control valve in a straight section of the piping of a length at least (20 or 25) d.

Depending upon the nature of the source of supply, flow control is classified into two basically different types: (a) with installation of the flow control valve in series (see Figs. 114 and 115), and (b) with parallel installation of the flow control valve (see Fig. 116 as well as flow control valve *6* in Fig. 194 of Vol. 1).

Systems (a) are more widely employed in machine tool engineering. Systems (b) are not suitable for circuits requiring rigidity or when several hydraulic motors are supplied from a centralized source (see p. 278).

In circuits (a) the flow-control valve can be installed either to control the flow of oil to the hydraulic reciprocating or rotary motor (Fig. 114, as well as flow control valve *6* in Fig. 194 of Vol. 1), or to control the rate of discharge from the hydraulic motor (Fig. 115, as well as flow control valve T_1 in Fig. 219 of Vol. 1). Circuits using the first method of control are called *metering-in circuits*; those using the second are called *metering-out circuits*.

A metering-out valve maintains a more uniform speed of the hydraulic motor; such circuits are usually applied for short travel and low hydraulic motor speeds ($v \leqslant 1$ m per min). However, the power consumption is higher in a metering-out circuit, in comparison with metering-in arrangements, because of the effect of the acting backpressure p_2.

Constant-displacement pumps are used with either type of flow control. In Soviet machine tools, these are either vane pumps, model Г12, or gear pumps, model Г11, depending upon the pressure required in the hydraulic system.

The efficiency of a system with flow controls is always less than that of one with variable-displacement controls. To compensate for this difference,

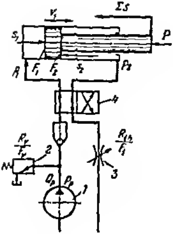


Fig. 115. Metering-out circuit for controlling piston speeds
1—pump; 2—relief valve; 3—flow control valve; 4—two-position directional valve

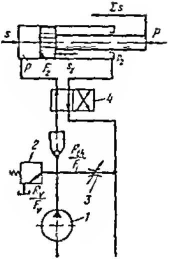


Fig. 116. Circuit with parallel installation of the flow control valve for controlling piston speeds
1—pump; 2—relief valve; 3—flow control valve; 4—two-position directional valve

the mechanical efficiency of constant displacement pumps should be higher, generally speaking, than that of variable displacement pumps. This cannot, however, be achieved in all cases. The maximum efficiency of a flow control system is about 0.65 or 0.67.

A certain part of the energy is expended and dispersed in the throttling process. For this reason, flow controls find application in hydraulic systems with a low power consumption, up to 3 or 3.5 kW. On the other hand, in cases when fast response or a high intensification factor is required, flow control systems are superior to all other possible solutions.

The pump delivery Q_p depends upon the maximum translatory or peripheral velocity of the hydraulic motor, the maximum volumetric losses in the system and a certain additional flow q_0 that continuously drains back to the tank but is necessary in order to maintain constant pressure in the system. This additional flow q_0 is established by adjustment of the spring of relief valve 2 (Figs. 114, 115 and 116).

The required pump delivery for a hydraulic cylinder is

$$Q_p = v_1'' F_1 + \sigma p_1 + q_0 \qquad (183)$$

while for a rotary hydraulic motor it is

$$Q_p = k_m n'' + \sigma p_1 + q_0 \qquad (184)$$

where v_1'' and n'' are the maximum velocity of the piston and maximum rotational speed of the hydraulic motor shaft, respectively.

There may be two reasons for nonuniform travel or rotation in a flow control system.

(a) One is the provision of a restrictor in the pressure or drain line, i.e. a sized orifice, through which the flow is $Q = C\sqrt{\Delta p}$; thus the stability of flow and consequent stability of speed depend upon the constancy of the pressure drop Δp. This is the so-called structural nonuniformity of the speed. In the circuit in Fig. 114, $\Delta p = p_p - p_1$; in Fig. 115, $\Delta p = p_2 - p_d$; and in Fig. 116, $\Delta p = p_p - p_d$.

(b) The other is kinematic compliance which depends upon the internal and external volumetric losses in the system.

Volumetric losses in the pump of flow control systems, such as those in Figs. 114, 115 and 116, only slightly affect the *kinematic* rigidity. The circuit with parallel installation of the flow control (Fig. 116) is the most sensitive to volumetric losses in the system and to changes in load. For this reason, it is mainly used in the servomechanisms of control facilities.

Consequently, nonuniformity of speed in flow control of a power load depends upon the structural nonuniformity.

Assuming that the no-load velocity v_0 (Fig. 114) is

$$v_0 = \frac{Q}{F_1} = \frac{c}{F_1}\sqrt{p_p - p_S} \tag{a}$$

where p_S is the pressure required to overcome the friction forces and other resistances (see p. 265), then the velocity v_1, at a load which develops the pressure p, is

$$v_1 = \frac{c}{F_1}\sqrt{p_p - (p + p_S)} \tag{b}$$

Next, using equation (161), we can determine the degree of structural nonuniformity. Thus

$$\chi = 1 - \sqrt{\frac{p_p - p_S - p}{p_p - p_S}} = 1 - \sqrt{1 - \frac{p}{p_p - p_S}} \tag{185}$$

The larger the ratio of p to the difference $p_p - p_S$, the larger the degree o structural nonuniformity will be.

With a reduction in area of the orifice in the flow control valve, the resistance in the pressure line increases, pressure p_1 in the cylinder decreases, as does the rate of input flow and velocity of the piston. An increase in the orifice area increases pressure p_1, leading to a corresponding increase in piston velocity.

If the orifice in the throttle is made with a special profile, the pressure p_1 in the cylinder and the piston velocity will vary in accordance with this

profile Therefore, in addition to control circuits, parallel installation of the throttle has found application in hydraulic devices requiring periodic changes in the pressure transmitted to the hydraulic motor (examples are clamping devices, friction clutches, certain types of presses, etc)

Since, at each point where the flow of fluid is divided, the algebraic sum of the inflow and outflow is equal to zero, we can write (Fig 116)

$$v_1F_1 = Q_p - \sigma p_p - \mu_l \sqrt{\frac{2g}{\gamma} p_p} F_l \quad (186)$$

where μ_l = discharge coefficient through restriction R

F_l = area of orifice through restriction R

Hence the piston velocity is

$$v_1 = \frac{1}{F_1}\left(Q_p - \sigma p_p - \mu_l \sqrt{\frac{2g}{\gamma} p_p} F_l\right) \quad (187)$$

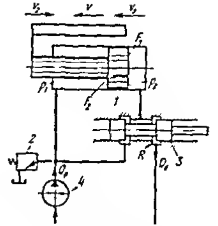


Fig 117. Differential flow control of piston speeds
1—metering orifice (diaphragm) 2—relief valve 3—throttle 4—pump

When the flow control valve orifice is entirely closed, i e at maximum piston velocity, we can write

$$v_1 = \frac{1}{F_1}(Q_p - \sigma p_p) \quad (188)$$

where σ is the coefficient of volumetric losses in the system.

It is evident from the last equation that this flow control method is sensitive to changes in pressure and to volumetric losses in the system

Using parallel installation of the flow control valve, the power transmitted to the hydraulic motor can be controlled in a wide range, from zero to the maximum value by varying the resistance of the valve Since only one restriction is used in the given circuit it is impossible to reverse the hydraulic motor with this resistance Reversal by means of resistances requires a valve with two resistances or two flow control valves (Fig 117)—the so-called differential flow control, in which the pressure line is connected to the cylinder end with the lesser piston area which should be $F_2 = 0.5 F_1$

The pressure is maintained stable by the provision of metering orifice *1*, its magnitude is regulated by relief valve *2* The size of metering orifice *1* must be calculated

If the resistance is regulated by means of restriction R, the pressure p_2 in the right end of the cylinder (in Fig 117) will vary correspondingly

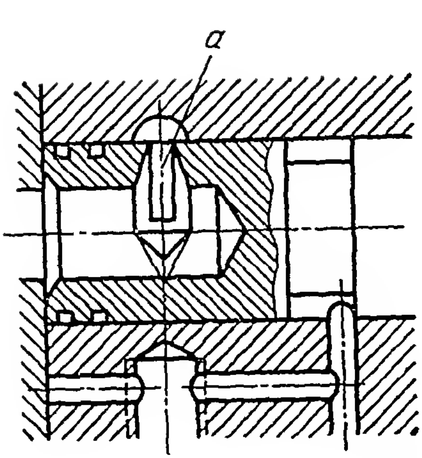


Fig. 118. Throttle with an orifice of diaphragm shape:
a—throttle

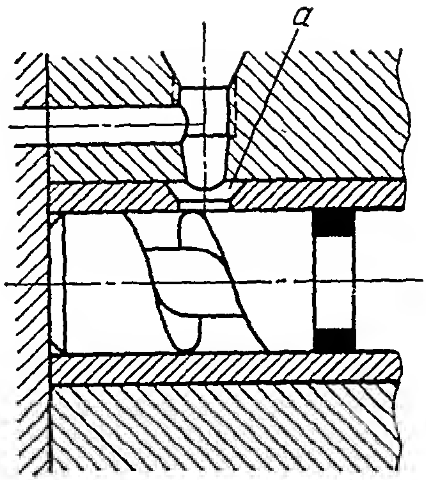


Fig. 119. Throttle in the form of a helical groove:
a—sleeve

For tentative and, in some cases, checking calculations, the maximum cross-sectional area of the throttle orifice can be taken equal to $F_i'' \cong 0.08\ d_0^2$, where d_0 is the inside diameter of the piping.

According to ENIMS, the flow through throttles with orifices of diaphragm shape (models Г77-11 and Г77-31) is

$$Q = 70 F_i \sqrt{\Delta p_{th}} \text{ litres per min} \tag{198}$$

where F_i = area of the throttle orifice, sq cm

$\Delta p_{th} \cong 0.34\ p_1$ = pressure drop in the throttle, kgf per sq cm.

Throttles of flow-control valves used in the hydraulic systems of machine tools must provide for an approximately constant flow at temperatures varying in a fairly wide range ($T = 15°$ to $50°$C). Tests show that satisfactory results are obtained when the length of the throttling slit is reduced. Therefore, the length of the orifice in new throttle and flow-control valves designed in ENIMS is equal to 0.1-0.2 mm. The flow through such an orifice approximates the ideal case of flow from an orifice with a flow factor of $\mu = 0.75$ to 0.78. The conductance of such an orifice varies with the pressure drop according to a parabolic function with an exponent of 0.5. The flow is almost constant in the above-mentioned temperature interval.

A throttle with an orifice of diaphragm shape is made by milling slit *a* in the hole of the stem (Fig. 118). Such an orifice can also be obtained as the mating of a helical groove on the stem with the longitudinal slot *a* (Fig. 119).

12-4. Stabilizing the Speed of the Hydraulic Motor

Structural nonuniformity of motion is inherent in flow control (see p. 292) and is due to variation in the load leading to variation in the pressure drop in the flow control valve.

Speeds can be stabilized by means of a reducing valve capable of maintaining an approximately constant pressure drop if it is properly connected to the flow control valve.

In metering-in speed control, the flow control and reducing valve are connected in parallel (Fig. 120), they are connected in series for metering-out control (Fig. 121).

When used for metering-in control (Fig. 120), the force p_{sp} exerted by spring *1* of the reducing valve compensates for the pressure drop in the flow control valve. Thus

$$P_{sp} = f_v(p_p - p_t) \tag{199}$$

where $f_v = \frac{\pi}{4}(d^2 - d_1^2)$ = effective area of spool *2* in the valve.

This equation does not take into account the reactive force developed as fluid flows over the conical tip *3* of the valve piston on its way to the tank through the outlet hole. This force depends upon the shape of the outlet hole. Deflector *4* is provided and spring *1* is additionally compressed to balance the valve piston.

As the load on the power piston is decreased, the pressure drops and the valve piston shifts to the right, increasing the flow of fluid draining to the tank. As a result, the fluid flow to the power cylinder is reduced. The pressure

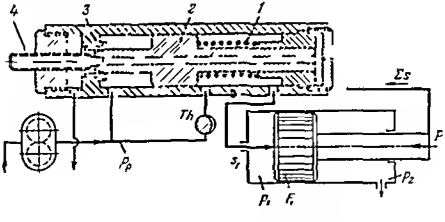


Fig. 120. Flow control valve with pressure compensator used in metering-in speed control

1—reducing valve spring; *2*—reducing valve piston; *3*—conical tip of valve piston; *4*—deflector

$p_1 = \frac{S_1}{F_1}$ (where $S_1 = P + \sum S_i$), in the end of the power cylinder, decreases somewhat, but the pressure drop $\Delta p_1 = p_p - p_1$ is restored. This circuit operates with a power consumption depending on the load. It is therefore a more economical arrangement than a metering-out circuit.

A hydraulic mechanism which is a combination of a reducing valve and a throttle is called a *pressure compensated flow-control valve* whose purpose is to stabilize the speed of the hydraulic motor in the circuit. The reducing valve reacts to each change in the force acting on the hydraulic motor, automatically adjusting its orifice for the corresponding fluid flow.

As an example, we can consider the circuit diagram for speed stabilization usually applied in Soviet machine tool engineering for metering-out controls (Fig. 121). Upon steady operation, the amount of fluid Q_2 passing out of the power cylinder and passing through orifice area f_x of the reducing valve, is equal to the amount of fluid Q_3 passing through orifice area F_{th} of flow control valve *3*, i.e. $Q_2 = Q_3$.

Upon a sudden increase in flow to the intermediate chamber *5* (for instance, when the tool runs out of the metal in planing), the input flow (Q_2) exceeds the output flow (Q_3). The surplus will raise valve piston *6*. The pressure in intermediate chamber *5* varies proportionally to the surplus (or decrease) in the amount of fluid. The following equation is therefore valid:

$$p_3 = p_3' + \Delta p_3$$

where p_3' = initial pressure in the intermediate chamber
Δp_3 = pressure increment in the same chamber.

Stabilization circuits have two elements: the controlled member (hydraulic motor) and the stabilizer (pressure compensated flow control valve), in which the role of the sensitive element is played by the area F_0' (Fig. 121) of the reducing valve piston. The member linking the sensitive element to the hydraulic motor is the flow control valve with an orifice area F_{th}. In its principle of operation, this stabilization system belongs to the category of static systems of direct control (each value of the controlled parameter corresponds to only a single position of the control element).

Representative of static control is its nonuniformity due to the absolute static error which is equal to the difference $\Delta m = m - m_0$ between the current position m of the reducing valve piston and its nominal position m_0. Hence, the nonuniformity of control is

$$\chi = \frac{m_{max} - m_{min}}{m_0}$$

If the controlled parameters remain unchanged during steady operation, i.e. when $Q_2 = Q_3$ (Fig. 121), then, as soon as this equality is upset, the parameters begin to change. The rate of this change is the higher, the less

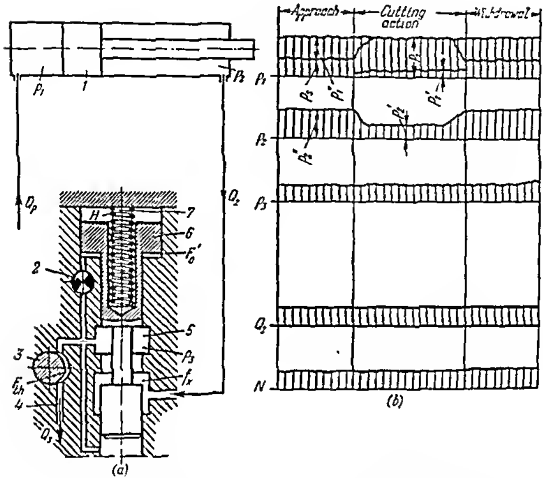


Fig. 121. Flow control valve with pressure compensator used in metering out speed control. 1—power piston; 2—damping device; 3—throttle; 4—port to tank; 5—intermediate chamber; 6—mushroom-type piston of reducing valve; 7—reducing valve spring

the volume of the hydraulic system, the greater the difference between the amounts of input and output fluid and the less the accumulating capacity of the hydraulic system.

Let us denote the flow to the control element by Q_2, the flow through the control element by Q_3, the controlled parameter (hydraulic motor speed in the given case) by v, and the parameter of the control element (travel of the reducing valve piston in the given case) by μ.

The rates of flow Q_2 and Q_3 are governed by the controlled parameter ν, and one of them is also dependent upon the parameter μ of the control element. Thus, for example, the rate of flow Q_2 (Fig. 121) is dependent upon parameters ν and μ, while Q_3 is governed only by parameter ν.

As a rule, stabilization circuits are single-capacity controlled members with two co-ordinates.

When equilibrium is upset, for instance at a decrease in the load, the input flow will exceed the through flow, i.e. $Q_2 > Q_3$, and then we can write for the transient process that

$$\vartheta \frac{d\nu}{dt} = Q_2(\nu, \mu) - Q_3(\nu) = \Omega \tag{200}$$

where Ω = index quantitatively representing the stabilization process, i.e. the flow of matter passing through the stabilized member (force pulse, oil flow to the hydraulic motor, torque pulse, etc.)

ϑ = index characterizing the inertia of the system.

For instance, in translatory motion, index ϑ represents the referred mass; in rotary or oscillating motion, it represents the referred moment of inertia.

For the sake of simplicity, the functions of the flow rates Q_2 and Q_3 in equation (200) are given as dependent upon two nonlinear parameters. Actually, they depend upon many nonlinear parameters.

Stabilization and control systems are investigated by making use of a linear model of the process; real systems of equations are converted into linear systems by expanding the component variables into a Taylor series according to the powers of their increments. The values of orders higher than the first are disregarded due to the small deviations from the steady state.

To linearize equation (200) we assume that

$$\nu = \nu_0 + \Delta\nu \quad \text{and} \quad \mu = \mu_0 + \Delta\mu$$

where the symbol Δ denotes an increment as usually.

Expanding Q_2 and Q_3 into a series we obtain

$$\left.\begin{aligned} Q_2 &= Q_{2.0} + \frac{\partial Q_2}{\partial \nu}\Delta\nu + \frac{\partial Q_2}{\partial \mu}\Delta\mu + \dots \\ Q_3 &= Q_{3.0} + \frac{\partial Q_3}{\partial \nu}\Delta\nu + \dots \end{aligned}\right\} \tag{a}$$

In these conditions, the subindex 0 indicates steady operation at which $Q_{2.0}(\nu, \mu) = Q_{3.0}(\nu)$. Taking this equation into consideration and substituting equations (a) into equation (200) we can write

$$\vartheta \frac{d(\Delta\nu)}{dt} + \left(\frac{\partial Q_3}{\partial \nu} - \frac{\partial Q_2}{\partial \nu}\right)\Delta\nu = \frac{\partial Q_2}{\partial \mu}\Delta\mu$$

Relative—dimensionless—values are introduced to facilitate the analysis of this equation. For this purpose all the increments of the variables are reduced to their nominal or maximum values. Then all the variables will be dimensionless and their coefficients will be either dimensionless or will have a time dimensionality. Thus let

$$\Delta v = v_0 \alpha \quad \text{and} \quad \Delta u = \mu_0 \beta$$

After substituting these values we have

$$\vartheta v_0 \frac{d}{dt}\left(\frac{\alpha}{\frac{\partial Q}{\partial \mu} l_0}\right) + \frac{\alpha v_0}{\mu_0 \frac{\partial Q}{\partial \mu}}\left(\frac{\partial Q_3}{\partial s} - \frac{\partial Q_2}{\partial v}\right) = \beta$$

Now we introduce the following notation

$$\vartheta \frac{v_0}{l_0} \frac{1}{\frac{\partial Q}{\partial \mu}} = T_0$$

and

$$\frac{v_0}{l_0 \frac{\partial Q_2}{\partial \mu}}\left(\frac{\partial Q_3}{\partial s} - \frac{\partial Q_2}{\partial v}\right) = \sigma$$

Then the preceding equation can be written as

$$T_0 \frac{d\alpha}{dt} + \sigma \alpha = \beta \tag{201}$$

where T_0 = time constant

σ — self regulation factor characterizing the property of the system as concerns stabilization

At $\sigma > 0$ the system requires no stabilization device since it possesses self regulating properties. At $\sigma < 0$ the controlled member is not capable of self regulation and consequently stabilization facilities are needed.

12-5 Synchronizing the Movements of Simultaneously Operating Hydraulic Motors

In some cases, as in hoisting and handling equipment, hydraulic presses, machine fixtures (for clamping at several points simultaneously) and in many other devices employed in various fields of engineering, it may be necessary to synchronize the movements of hydraulic motors.

In practice it can be observed that the synchronism of hydraulic motors is upset from the very moment they start due to the difference in the static friction forces developed in various parts of the motors. Synchronism is not restored during subsequent motion of these hydraulic motors.

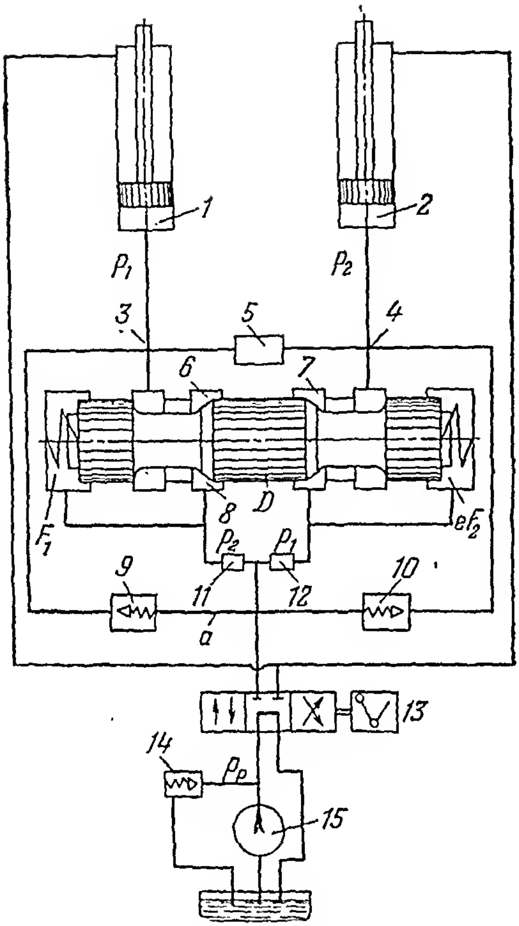


Fig. 122. Circuit for synchronizing the speeds of two hydraulic motors (power pistons):

1 and *2*—power pistons; *15*—pump; *14*—relief valve; *D*—divider plunger f[illegible] sures p_1 and p_2; *6*, *7* and *8*—[illegible] of the plunger; *11* and *12* [illegible] (diaphragms); *9* and *10*—check valves; *13*—three-position directional valve; *5*—connecting throttle; F_1 and F_2—areas of the flow-divider plunger

Various circuits are used to synchronize the movements of two or more hydraulic motors (power cylinders or rotary motors). Most widespread are circuits with flow-divider valves.

A necessary condition for more or less accurate synchronization is that the hydraulic motors must have the same characteristics: equal piston diameters for power cylinders and equal displacement of the rotary hydraulic motors for rotary drives.

Throttle-type flow-divider valves (Fig. 122) have found wide application for synchronizing circuits. The pump output Q_p is delivered at constant pressure p_p into chamber *a* and further, through two parallel lines, through resistances $11R_1$ and $12R_2$, the oil passes into chambers F_1 and F_2. At rates of flow Q_1 and Q_2 , respectively, the two streams of oil pass through throttling orifices *6* and *7*, to the ports of lines *3* and *4* which are connected to the two hydraulic motors.

If the loads on the hydraulic motors are the same, the pressures in lines *3* and *4* are equal, and the divider plunger *D* is in its central (neutral) position, in which it divides the pump output equally between the two hydraulic motors. If the pressure in line *4* increases due to an increase in load, the pressure also increases in chamber *e* so that divider plunger *D* shifts to the left. At this, throttling orifice *7* increases and orifice *6* is reduced. The pressures in the chambers will continue to change until the pressure drops are equalized, thereby equalizing the rates of flow in lines *3-1* and *4-2*.

When the pressure in line *3-1* becomes equal to that in line *4-2*, hydraulic motor *2* stops, and divider plunger *D* shifts to the right, closing the throttling orifice to some extent. Then hydraulic motor *1* also stops.

The ratio of the divided rates of flow should not exceed 2 : 1, and even in this limiting case, an error in synchronism not over 4 per cent can be guaranteed.

An increase in flow through the dividing valve above the indicated limit leads to a reduction in the efficiency of the system, while a decrease in flow, below a definite value, may sharply increase the error in flow dividing.

The response time of the flow divider valve has a marked effect on the accuracy with which it operates. This time depends upon the effective area, weight and travel of divider plunger *D*. The less this travel, the more accurately the flow will be divided.

At high pressures in the system, this accuracy also depends upon the compressibility of the oil and the volumetric losses of the pump. It is therefore advisable in such cases to install the divider valve as close as possible to the hydraulic motors.

The action of a flow divider valve is somewhat similar to that of a Wheatstone bridge. Oil flow in the dividing lines is

$$p_p - p_1 = Q_1^2 R_1 \quad \text{and} \quad p_p - p_2 = Q_2^2 R_2$$

Since, according to the conditions, $p_1 = p_2$, the right sides can be equated. Therefore

$$\frac{Q_1}{Q_2} = \sqrt{\frac{R_2}{R_1}}$$

12-6. Hydraulic Power Amplifiers and Tracer-Controlled Systems

Hydraulic amplifiers are extensively employed in systems for the automatic control of the operation of a great variety of machines. These devices are hydraulic mechanisms of small overall size which are characterized by high amplification coefficients, fast response and high dependability.

Most commonly applied in machine tool design are spool type hydraulic amplifiers, distinguished for their simple construction, small size and high sensitivity. Fluidic control of hydraulic amplifiers finds almost no application in machine tools.

The block diagram of a hydraulic amplifier (Fig. 123) includes a measuring device (sensitive element) whose purpose is to compare (or measure) two positions: the required position X fed in by the input element (*IE*), and the actual position Y, reached by the hydraulic motor (operative member) *2*

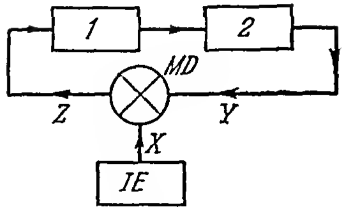


Fig. 123. Block diagram of a hydraulic amplifier

The difference $X - Y = Z$ is the tracing error, or mismatch. The error signal is fed into converter *1*. The measuring device (MD) has two inputs Y and X and one output Z.

In single-co-ordinate systems of tracer-controlled machine tools, the input element is the template T (Fig. 124*a*), the sensitive element is the tracer valve V, the converter (or amplifying unit) is piston *1*, while cylinder *2*, to which the tool is rigidly secured (it is shown mounted directly on the cylinder in the diagram), is the operative member in this case.

The output of the pump is delivered into cylinder end F_1 and simultaneously, through orifice F_x of the valve, to end F_2 with a backpressure p_2. The value of p_2 is limited by the spring of valve *4*.

The level of pressure p_2 is determined with the hydraulic motor stationary and at a certain preliminary opening m_0 of valve *3* at which the pressure drop between the ends of the hydraulic motor is $\Delta p = 0$. In this case, $p_2 = p_1 \frac{F_1}{F_2}$ and, since $F_1 = 0.5\, F_2$ as a rule, then $p_2 = 0.5\, p_1$.

If valve piston *3* is displaced from position m_0 to the right, cylinder *2* will also travel to the right with a velocity of v_1, and oil from the opposite end will be forced out at constant pressure p_2 back to the tank.

Pressure p_1 in the right end of cylinder *2* will vary continuously in accordance with the pressure drop Δp in the valve. Therefore, $p_1 = p_2 + \Delta p$.

If valve piston *3* is shifted to the left (on the diagram) from position m_0, the pressure drop Δp will be reduced, and cylinder *2* will start to travel to the left in the direction of v_2.

We can write the flow equations for a certain current area F_x of the orifice in valve *3*. Thus for junction point *a* (Fig. 124*b*)

$$Q = v_1 F_1 + q_x \qquad \text{(a)}$$

and for junction point *b*

$$Q_0 = v_1 F_2 + q_x \qquad \text{(b)}$$

From these two equations we find the required pump delivery to be

$$Q = Q_0 - v_1 (F_2 - F_1) \qquad \text{(c)}$$

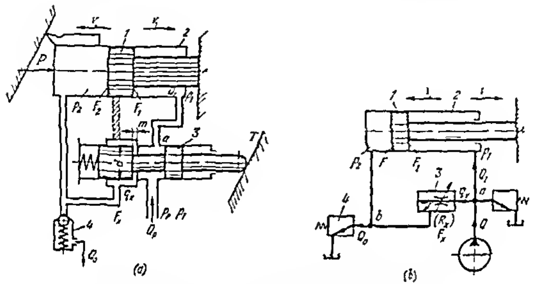


Fig. 124. Single-co-ordinate parallel tracer control
1—piston 2—power cylinder 3—tracer valve 4—backpressure valve

Using the concept of conductance (see p. 278) we determine the rates of flow

$$q_x = s_x \sqrt{p_1 - p_2}$$

where s_x = conductance of the orifice opening area in the valve *3* and

$$Q_0 = s_0 \sqrt{p_2}$$

where s_0 = conductance of the opening in valve *4* (Fig. 124).

Substituting these flow values into equation (*b*) we obtain

$$s_x \sqrt{p_1 - p_2} = s_0 \sqrt{p_2} - v_1 F_2$$

from which the current conductance of the orifice area in the valve *3* for a tracing speed of v_1 is

$$s_x = s_0 \frac{1}{\sqrt{\frac{p_1}{p_2} - 1}} - \frac{v_1}{\sqrt{p_1 - p_2}} F_2 \qquad \text{(d)}$$

Maximum conductance occurs at a tracing speed of $v_1 = 0$. This conductance corresponds to the preliminary opening of the tracer valve, i.e. to its position m_0. Hence

$$F_{x\,0} = F_0 \frac{s_x}{s_0} \frac{\mu_0}{\mu} = F_0 \frac{\mu_0}{\mu} \frac{1}{\sqrt{\frac{p_1}{p_2} - 1}} \qquad \text{(e)}$$

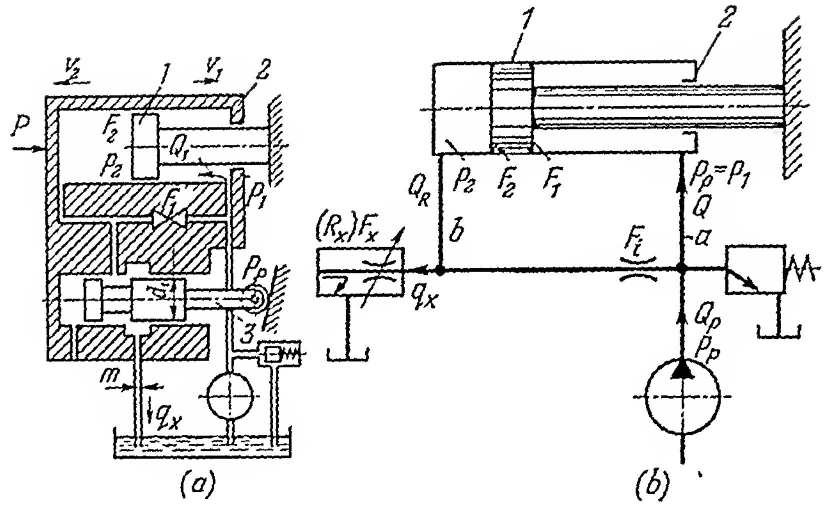


Fig. 125. Single-co-ordinate tracer-control system with constant input pressure p_p: 1—power piston; 2—power cylinder; 3—tracer valve

The maximum tracing speed will obviously be at $s_x = 0$. In this case, it follows from equation (d) that

$$v_1 = \frac{s_0}{F_2}\sqrt{p_2} \tag{f}$$

Knowing the conductance of various openings, and making use of equation (157), the geometrical dimensions of the openings can be determined. The flow equations are derived in the same way for the direction along v_2 (Fig. 124); all the necessary data are determined likewise.

A parallel tracing circuit of this type operates with variable power consumption and is therefore more economical than a series tracing circuit and moreover, the oil is heated less. However, the parallel circuit is less stable, due to its structure, than other circuits and is rarely used in machine tools.

Constant-pressure tracing systems (Fig. 125*a* and *b*) are widely employed in machine tool design thanks to their high rigidity and sensitivity.

Tracer valve *3* controls the oil discharged from the larger end F_2 of the cylinder. In the opposite end F_1, the pressure $p_1 = p_p$ is maintained constant. The two ends F_1 and F_2 of the cylinder are connected by orifice F_i which actually passes through the piston.

Displacement of the valve piston changes the pressure in end F_2 of the cylinder. This, in turn, leads to motion of cylinder *2* and the cutting tool secured to it.

The neutral position of the valve piston, i.e. the position in which the cylinder is stationary, is determined from the condition that $p_1F_1 = p_2F_2$.

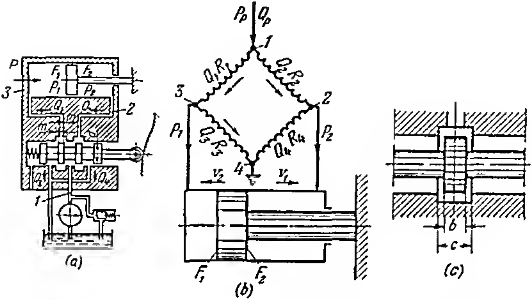


Fig 126 Diagram of a single co-ordinate tracing system using a four-edge tracer valve

Hence

$$\frac{F_2}{F_1}=\frac{1}{1-\frac{\Delta p}{p_1}}$$

where $\Delta p = p_1 - p_2$.

If the valve piston is shifted to the left from its neutral position, the cylinder will travel to the left at a speed v_2 because of the difference in piston areas on the two sides

Using the diagram (Fig 125b) and the data given on p 293, there should be no difficulty setting up the flow equations and then determining all the required design data

This is the simplest type of tracing control circuit It is usually employed in cases when it is not necessary to reproduce the contour of the template with very high accuracy when there are no sharply alternating loads, and when the tracing error is relatively small in comparison with other errors introduced in the system by other mechanisms and units of the machine tool

Tracing circuits with a four-edge valve (Fig 126a, b and c) are used when higher accuracy of contour reproduction is required, as well as high rigidity.

While the operation of single-edge tracer valves is based upon the difference in areas on the two sides of the power piston (see, for instance, Figs. 124 and 125), a four-edge valve does not require different areas. Therefore, the latter type of tracer valve can be employed in rotary tracing drives as well.

Such a four-edge tracer valve (Fig. 126*b* as well as Fig. 167 of Vol. 1) resembles a Wheatstone bridge to some extent in its operation. It controls motion of the hydraulic motor by varying the pressures at the ends F_1 and F_2 of the cylinder. In the central (neutral) position of the valve piston, the orifices of the valve are equal. Therefore, $\frac{R_1}{R_2} = \frac{R_3}{R_4}$ and the pressures on the two sides of the power piston are also equal.

As the valve piston shifts to the right, resistances R_1 and R_4 are increased and resistances R_2 and R_3 are reduced. As a result, pressure p_1 is reduced and pressure p_2 is increased.

It has been shown that a valve piston movement of only 0.025 mm can produce a difference between pressures p_1 and p_2 of as much as 60 per cent of the applied pump pressure.

The clearance *c-b* (Fig. 126*c*) usually ranges from 16 to 20 microns. Thus, if the radial clearance between the valve piston and body does not exceed 10 microns, flow through the radial clearance can be neglected.

The speed of the piston depends upon the difference between the rates of flow to the junction points. Hence, the speed $v_2 = \frac{Q_2 - Q_4}{F_2}$. Consequently, at $v_2 = 0$, $Q_2 = Q_4$. This enables an equation to be set up for the balanced position of the valve spool (Fig. 126*b*):

$$p_1 - p_2 = p_p - 2p_2$$

or

$$p_p = p_1 + p_2 \tag{202}$$

The backpressure p_2 upon piston travel in the direction of v_1 is

$$p_2 = \frac{F_2^2 v_1^2 \gamma}{2g\mu^2 f^2}$$

If this value is substituted into equation (202), and it is assumed that the orifice of the valve has the shape of a ring, i.e. $f = \pi\, dm$, the required pump pressure will be

$$p_p = p_1 + \frac{F_2^2 v_1^2 \gamma}{2g\mu^2 (\pi\, dm)^2} \tag{203}$$

The pressure p_l in the cylinder is determined by equation (121) The flow equations for the arms of the bridge (Fig 126b) are

$$Q_p = Q_1 + Q_2 \text{ and } v_1 F_1 = Q_1 - Q_3$$
$$Q_p = Q_3 + Q_4 \text{ and } v_2 F_2 = Q_2 - Q_4$$

The component rates of flow can be determined on the basis of the conductances of the bridge arms

$$Q_1 = s_1 \sqrt{p_p - p_1} \quad Q_2 = s_2 \sqrt{p_p - p_2} \quad Q_3 = s_3 \sqrt{p_1} \quad Q_4 = s_4 \sqrt{p_2}$$

Tracer-controlled servomechanism systems are closed-loop control systems with directed connections that continuously compare the specified position X with the actual position Y of the controlled member This comparison is carried out by means of negative feedback, operating so that the output co ordinate Y is fed into the measuring device (see Fig 123)

Since the servo valve does not perform any work, it does not transmit energy, but only controls the operative member Consequently, it is not the power of signal X of the input element that is of prime importance, but only the information about this signal

Continuous-action tracing systems have found widest application in machine tools In these systems the control signal appears simultaneously with the mismatch Evidently, the less the mismatch, the more accurate the tracing action will be

A hydraulic amplifier is characterized by the coefficient of power amplification K_p which is defined as the ratio of the output force P to the input effort C, i e to the force exerted on the valve spool. In many cases, values of $K_p > 3 \times 10^3$ are obtained in hydraulic amplifiers

The quality of a hydraulic amplifier is assessed by the quality factor $Qt = \frac{K_p}{T}$, where the time constant $T = 0.005$ to 0.01 sec

In rotary tracing drives the quality factor is the ratio $Qt = \frac{\omega_y}{\theta}$, where ω_y is the steady idle angular velocity of the output shaft, and θ is the mismatch angle

The quality factor of translatory and rotary tracer controlled drives is assigned so that the velocity and static errors of the system are within the specified values In addition, the system must have a sufficiently high margin of stability

In some cases, the quality factor Qt is selected according to the frequency response curves of the system

Three types of valves are used in tracer controlled systems The first type includes valves with underlap (Fig 126c) in which there is always a certain oil flow when the valve is in the neutral position Such valves are highly sensitive but consume slightly more power for the nonproductive flow of oil

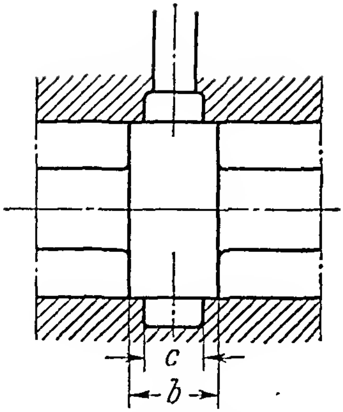


Fig. 127. Tracer-control valve with overlap blocking off the pressure line

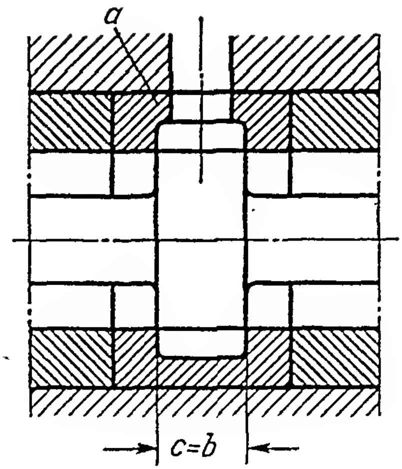


Fig. 128. Tracer-control valve with zero lap blocking off the pressure line

in the neutral position of the valve. Valves of the second type have overlap (Fig. 127) so that in the neutral position the input port is closed and flow from the pressure line is drained back to the tank through the relief valve. These tracer valves are used when it is necessary to switch on the hydraulic motor periodically, as in control units, step motors and selectors, etc. These valves have a dead zone, determined by the difference $\Delta = b - c$; this promotes an increase of the time constant T.

Efforts have been made to raise the rapid response and sensitivity by employing the third type of valves, the so-called zero-lap, or ideal, valves (Fig. 128) in which $b - c = 0$. Such valves, however, require high accuracy in manufacturing and installation. An additional part—a sleeve made up of separate rings a has been provided to facilitate manufacture.

In the USSR tracer valves are usually made of alloy steel, grade X1H or 18X2H4BA, which is subsequently carburized and hardened to 59-62 R_C. If the valve has a sleeve, it is made of steel, grade 40XHMA or XΓ, heat-treated to the same hardness. The end faces of the rings must be precisely ground, and be strictly square to the axis. The face runout of the ends should be within 3 microns.

Following heat treatment and hardness inspection, the sections of the valve spool are to be ground to the 10th or 11th class of surface finish according to USSR Std GOST 2789-59. The permissible out-of-roundness and taper should be within 2 to 5 microns. No breaking or rounding-over of the edges is permitted. The valve spools are ground individually to their bores to obtain a clearance of 5 to 12 microns, in accordance with the valve diameter.

12-7. One- and Two-Dimensional Controlled Tracer Systems

Hydraulic tracing slides are attachments widely used in the metal-working industries. They are mounted directly on engine lathes and are used for turning stepped, tapered or contoured surfaces of revolution.

The best known of these slides in the Soviet Union is model KCT-1 operating on the circuit shown in Fig. 125. This, and other tracing slides for lathes operate on the one-dimensional controlled tracing principle, i.e. the velocity s of the Input motion is constant and this motion is driven by the lead screw or feed rod of the lathe. The velocity v_t of the tracing motion of the cutting tool is obtained by geometrically adding the carriage velocity s and the velocity v_1 of the piston (Fig. 129*a*, *b* and *c*). For a section being traced (Fig 129*a* and *c*), we can write in accordance with the law of sines that

$$\frac{s}{\sin(\beta-\alpha)}=\frac{v_1}{\sin\alpha}=\frac{v_t}{\sin\beta}$$

where β is the angle between the tracing slide and the main slide of the machine.

Making use of these equations, and substituting the values of angles β and α, we can find v_t in accordance with s, α and β.

It is evident from these equations that in the general case of turning a contoured surface of revolution, i e when the velocity of the Input motion is constant but the angles β and α are variable, the obtained resultant velocity v_t can be calculated as

$$v_t=\frac{s\sin\beta}{\sin(\alpha-\beta)}$$

This indicates that at constant velocity s, velocity $v_t = f(\alpha, \beta)$ will be variable, in some cases in a wide range This is one drawback of this type of tracer-controlled machining

The angle α of the rising slope of the contour (Fig. 129*a*) may vary from $\alpha = 0$, in turning a cylinder, to $\alpha = 90$

In turning a descending slope of the contour (Fig 129*b*), the angle $-\alpha$ is determined from the condition of constant contact of the tracer stylus with the template profile, and on the basis of the nose angle ε of the tool. The nose radius of the tool and the tip radius of the stylus are made the same These circumstances, as well as the necessity for roughing and finishing workpieces of different hardness, are the reasons why the stylus is usually of the interchangeable type

The accuracy of the reproduced contour depends upon the accuracy of the template, the agreement between the radii of the tool nose and stylus tip, the mismatch values, elastic strain of the machine units and certain other factors

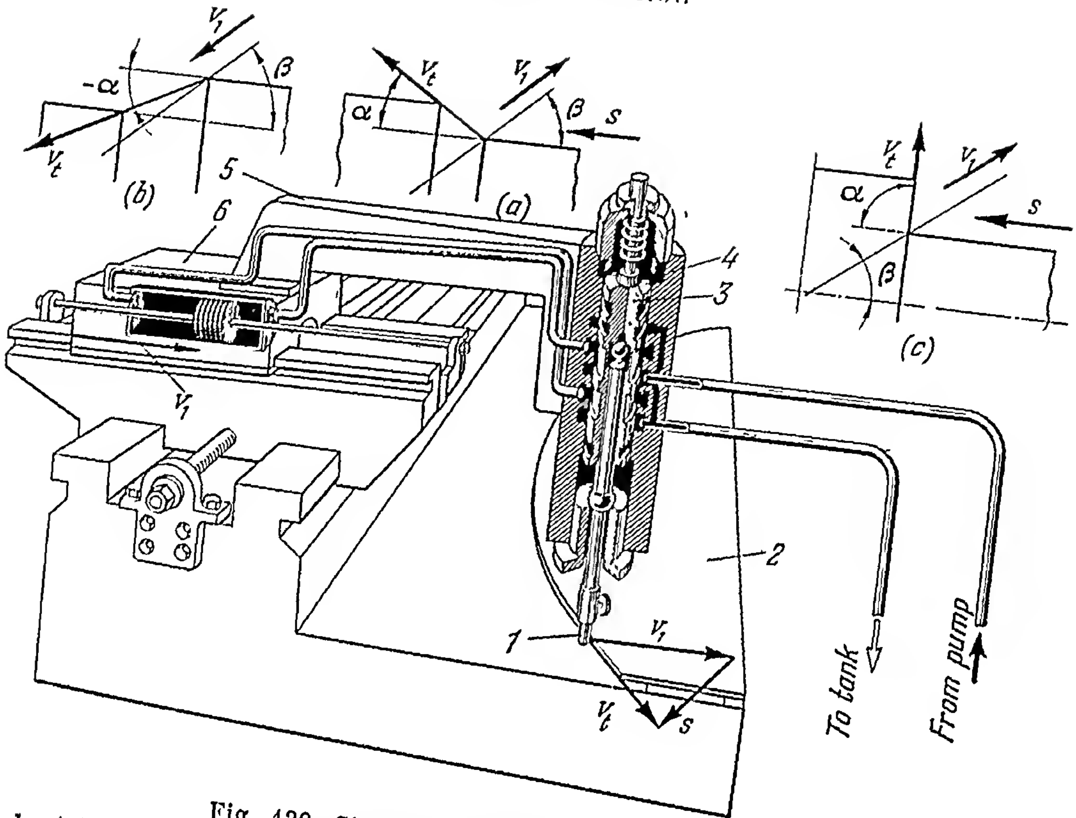


Fig. 129. Single-co-ordinate tracing system:
1—stylus; 2—template; 3—tracer valve; 4—valve housing; 5—bracket; 6—toolholder

Practically attainable diametrical accuracy in tracer-controlled turning ranges from 0.05 to 0.1 mm for the corresponding mismatch values from 0.02 to 0.04 mm.

Tolerances on the tool nose and stylus tip radii should not exceed 0.1 mm, being +0.1 mm for the tool and —0.1 mm for the stylus. The tool should be set up to the line of centres of the lathe with an accuracy within ±0.25 mm.

If the constant resistance F_t in the tracer-controlled circuits (see Figs. 117 and 125) is replaced by a variable resistance, using valve model БГ54 for this purpose, the transient processes in the system will be of shorter duration and, as a result, the accuracy of reproduction will be increased. In addition, the most proper pressure drop between the ends of the power cylinder can be set by adjusting the spring of this valve.

A constant tracing velocity can be maintained in two-dimensional, independent, mutually perpendicular tracer control (Fig. 130) in which the tool is fed simultaneously along two mutually perpendicular co-ordinate axes

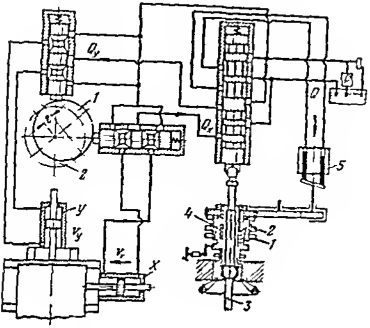


Fig 130 Two-dimensional tracer-controlled system

The principal elements of a two dimensional tracing system are two hydra[illegible] four edge tracer valves and a sine-cosine [illegible] whose profile provides for oil flow rates [illegible]tional to the sine and cosine of the angle of rotation of the distributor

The control valve rod is linked through a pivot joint to stylus *3* Mounted on taper sleeve *4* of this valve is distributing disk *1* which is driven by axial piston rotary hydraulic motor *5*

Any change in the eccentricity *e* in reference to the axis of rotation of sleeve *4* changes the flow of oil to the hydraulic cylinders

To maintain a constant resultant (tracing) velocity v_t it is necessary that velocities v_x and v_y along the co-ordinate axes are continuously changed One should vary proportionally to the cosine and the other proportionally to the sine of the angle of distributor rotation Thus $v_x = v \sin \alpha$ and $v_y = v \cos \alpha$ As a result v_t will be proportional to the eccentricity *e* and its direction will approximately coincide with that of the eccentricity

The contour of the workpiece obtained in machining as a result of geometrically adding velocities v_x and v_y has practically a very small error

The delivery Q_p of the pump is distributed among three directions in parallel: Q_1, Q_x and Q_y (Fig. 130).

The reproduction error is dependent upon the errors in the kinematic chains of the machine tool, errors of the fixture and cutting tool, and errors in the tracing system proper.

Tool errors depend, for instance, upon the lack of coincidence between the dimensions of the tool nose and stylus tip, and their positions in reference to the workpiece and template.

Kinematic errors are due mainly to backlash and elastic strain in the kinematic chains.

Tracing errors depend upon dynamic factors such as the time required for the damping of transient processes, certain additional deflections of the cutting tool due to chance factors, and the stationary error of the system.

Rotary hydraulic motors, model MГ15 (see Fig. 93), can be used in tracer-controlled servo systems as drives to the machine tool carriages and saddles by connecting the lead screw to the motor shaft. A construction of this type is especially efficient in the tracer-controlled machining of large work.

In modifying engine lathes so that they are capable of performing contour-turning operations to a template, the use of a rotary hydraulic motor will greatly simplify matters and exclude the need for making additional parts and units for the lathes.

Rotary hydraulic motors have much higher inertia than power pistons. It is not advisable, therefore, to use rotary motors in the tracing circuits of high-speed machines.

The application of rotary hydraulic motor, model MГ15, for turning the table of a heavy vertical boring mill to an angle, is shown schematically in Fig. 131.

This circuit includes pump *1* of either the vane or gear type, depending upon the required pressure which is set by adjusting the spring of relief valve *2*, and servo valve *3* whose rod is linked rigidly to feedback nut *4*. Gear *6*, keyed on the shaft of the rotary hydraulic motor, is in constant mesh with the toothed rim of nut *4* and simultaneously meshes with gear *7* of the table.

The command for angular motion (indexing) of the table is given by turning handwheel *5*. This traverses nut *4* along the screw of handwheel *5*, shifting the spool of valve *3* and admitting oil to the rotary hydraulic motor. The motor rotates gear *7* together with the table, and simultaneously rotates feedback nut *4*.

This nut is now traversed in the opposite direction along the screw; it shifts the valve piston back to the neutral position which is reached when the table has been turned to the preset angle.

The time required for the hydraulic motor to reach full speed is

$$t_l = \frac{J_l \omega}{M_l}$$

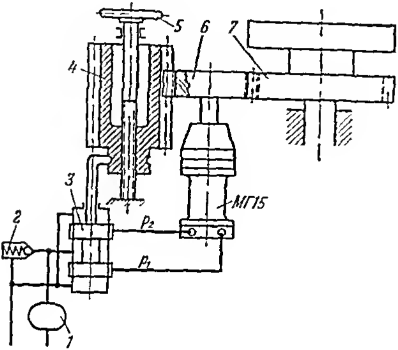


Fig 131 Servo system controlling the angle of rotation of a rotary hydraulic motor

where ω = steady angular velocity of the rotary hydraulic motor
J_l = referred moment of inertia
M_l = torque developed on the shaft of the hydraulic motor

In the given arrangement the referred moment of inertia is

$$J_l = J_m + J_6\left(\frac{\omega_1}{\omega}\right)^2 + J_7\left(\frac{\omega_2}{\omega}\right)^2$$

where J_6 and ω_1 = moment of inertia of gear 6, and its angular velocity, respectively
J_7 and ω_2 = moment of inertia of gear 7 and the table, and their angular velocity respectively
J_m = moment of inertia of the hydraulic motor

The torque developed on the motor shaft is

$$M_l = \frac{1}{2\pi} K_2 (p_1 - p_2)$$

where K_2 is the displacement of the rotary hydraulic motor per revolution

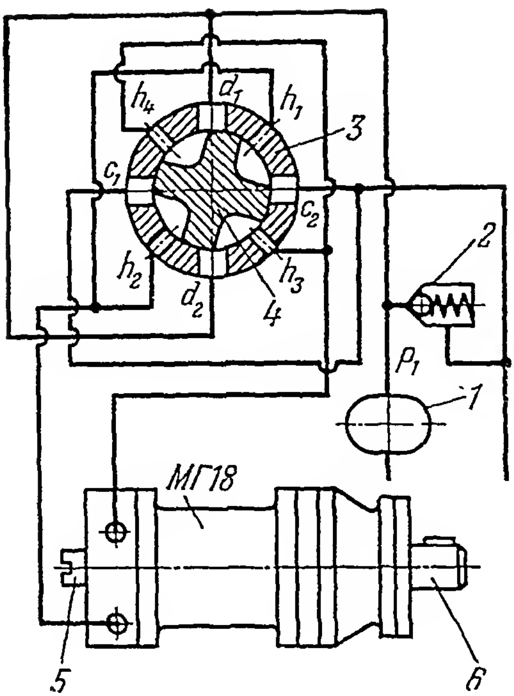


Fig. 132. Diagram of a hydraulic torque amplifier

If a rotary type of servo valve, such as *3* and *4* in Fig. 132, is used in the arrangement of Fig. 131 in place of servo valve *3*, a model МГ15 rotary hydraulic motor can be employed as a hydraulic torque amplifier. This type of circuit has been widely applied in synchronizing devices with input and output shafts (a sort of hydraulic synchrotie) which provide, whenever necessary, an increase in torque on the output shaft. In Fig. 132, *1* and *2* are the pump and the relief valve, respectively.

Spool *4* of the servo valve (Fig. 132) is connected with a transmitter of input signals of any type which is capable of rotating the spool at constant or variable speed. In most cases, it proves convenient to connect the spool to a low-power step motor. Body *3* of the valve is linked through a constant-velocity coupling (one with no backlash) to the end of shaft *5* of the hydraulic motor. In the model МГ15 motor, provision has been made for such connections (see Fig. 93). The second end *6* of the shaft is linked to the load.

The МГ18 torque amplifier has been developed on the basis of the МГ15 motor.

The pump pressure p_1 is applied to ports d_1 and d_2, while ports c_1 and c_2 of the valve body are connected to the tank. With the model МГ18 hydraulic amplifier, the valve is connected through the two pairs of ports h_1-h_2 and h_3-h_4.

The input angle of rotation is determined by the angular setting of valve spool *4*. In turning, the spool admits oil, under pressure, into the hydraulic amplifier. At this, the amplifier shaft, and the feedback shaft *5* begin to rotate. At the same time that the load is turned, body *3* of the valve is turned with it in the same direction until the preset angle of rotation is reached. If the input shaft is coupled to a low-power step motor, the power rating of the motor can be found by first determining the load, equal in this case to

$$M_{n\,0} = M'_n + M''_n + M'''_n \qquad (204)$$

where M_n' = torque required to overcome the inertia of spool *4* and the input shaft

M_n'' = torque required to overcome the reactive force due to the reaction of the outflowing stream of oil

M_n''' = torque required to overcome friction that is dependent upon the pressure in the system and the size of the hydraulic amplifier

The torque M_n' is determined from the equation

$$M_n' = J_1 \frac{d^2\varphi}{dt^2}$$

where J_1 is the moment of inertia of spool *4* and the input shaft

The moment of inertia J_{in} of the input shaft depends upon the size of the hydraulic amplifier. Thus for model МГ18-12 $J_{in} = 0.2$ kgf-cm sec², for model МГ18-13 $J_{in} = 0.6$ kgf-cm sec², and for model МГ18-15 $J_{in} = 0.24$ kgf-cm sec²

The torque M_n'' required to overcome the reactive force is determined from the formula (according to data of ENIMS)

$$M_n'' = 1.64KQp^{0.5} \text{ kgf-cm}$$

where Q = oil delivery to the hydraulic amplifier, litres per min

K = experimental factor depending upon the size of the hydraulic amplifier

Model	МГ18-12	МГ18-13	МГ18-15
K	0.03	[illegible]	0.065

The torque M_n''' required to overcome friction is listed in Table 5

TABLE 5

Model of hydraulic amplifier	Torque M_n''' kgf-cm in accordance with the pressure in the system, kgf per sq cm		
	20	40	50
МГ18-12	0.07	0.07	0.07
МГ18-13	0.2	0.25	0.3
МГ18-15	0.3	0.5	0.6

The torque M_{op} developed on the output shaft of the hydraulic amplifier is

$$M_{op} = M_{ld} + M_f + M_j$$

where M_{ld} = torque of the load, for instance, from the cutting force
M_f = friction torque of the mechanism driven by the amplifier
M_j = torque required to overcome the inertia of the driven mechanism.

$$M_j = (J + J_m)\frac{d^2\omega}{dt^2}$$

where J is the moment of inertia of the output shaft of the hydraulic amplifier (for model MГ18-12, $J = 0.12$ kgf-cm-sec^2; for model MГ18-13, $J = 0.38$ kgf-cm-sec^2, and for model MГ18-15, $J = 0.26$ kgf-cm-sec^2)
J_m = moment of inertia of the driven mechanism
ω = angular velocity of the output shaft
$\frac{d^2\omega}{dt^2}$ = angular acceleration, sec^{-2}, of the output shaft.

The hydraulic torque amplifier permits up to 1,600 pulses per sec or up to 1,200 pulses with an amplitude of 1.5° per pulse. This is substantially higher than the permissible number of pulses for low-power electric motors. The large available number of pulses enables the pitch to be reduced, thereby raising the accuracy of angular movements. The advantageous features enable hydraulic amplifiers to be used in numerically controlled machine tools in conjunction with low-power step motors.

It is well known that automatic transfer machines and the integrated automation of manufacturing processes are being more and more extensively employed in large-lot and mass production.

Small-lot production requires machine tool automation of a type permitting rapid change-overs from workpiece to workpiece. In this respect, most promising are the application of various tracer-controlled devices and systems of numerical controls.

As mentioned previously, hydraulic tracer-controlled slides are installed on various machine tools of the lathe group. These attachments turn the work either to a template or to a model. The latter is a master part turned without the use of a template. Hydraulic tracer-controlled slides have also been used on machine tools of other groups, for example, planers.

Tracer-controlled slides are installed either in front of the lathe (Figs. 133 and 134) or at the rear (Fig. 135), at an angle from 45° to 60° in reference to the line of centres. This arrangement of the slide permits shoulders to be turned at an angle of 90° to the work axis.

If the tracing slide is arranged at the rear, the versatility of the lathe is in no way impaired since the regular square turret can be used at the front.

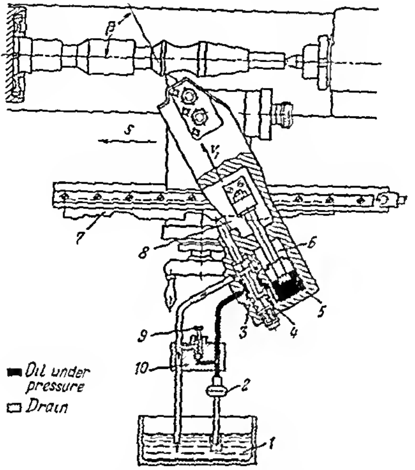


Fig 133 Hydraulic tracer-controlled slide for front installation

1—tank 2—pump 3—housing of tracing device 4—tracer valve 5—front end of power cylinder 6—rear end of power cylinder 7—template 8—stylus 9—adjusting screw 10—relief valve

of the work This enables a model workpiece to be turned in the conventional manner for subsequent use as a template

The template is secured either below or above the slide Special additional brackets will be needed in the second case this may require modifications in the construction of the lathe

If the tracing slide is installed at the rear the cutting zone is accessible to observation and the template is out of the way of the chips

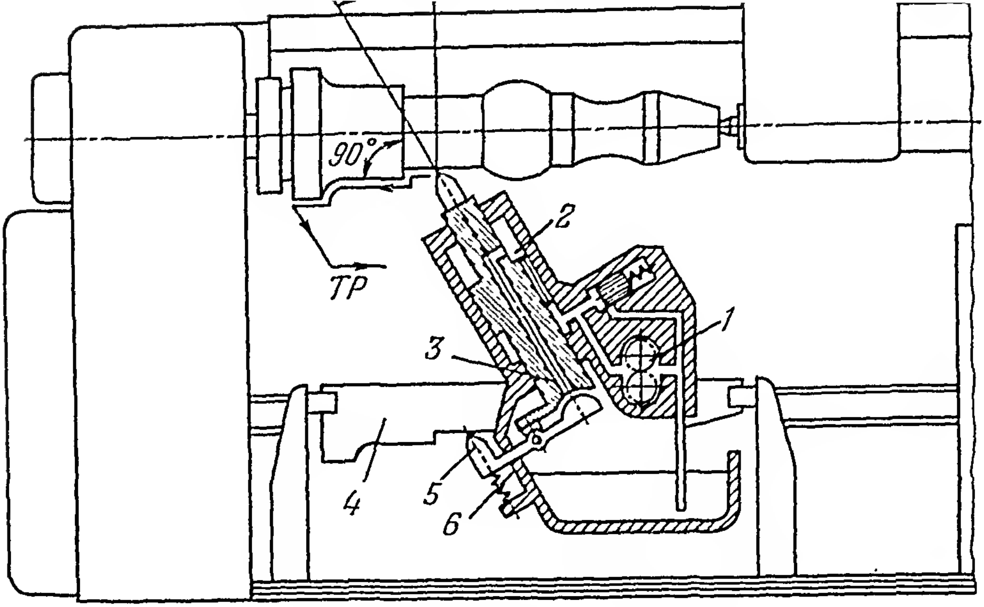


Fig. 134. Hydraulic tracer-controlled slide for front installation:
1—gear pump; *2*—power piston; *3*—tracer valve; *4*—template; *5*—stylus; *6*—stylus spring

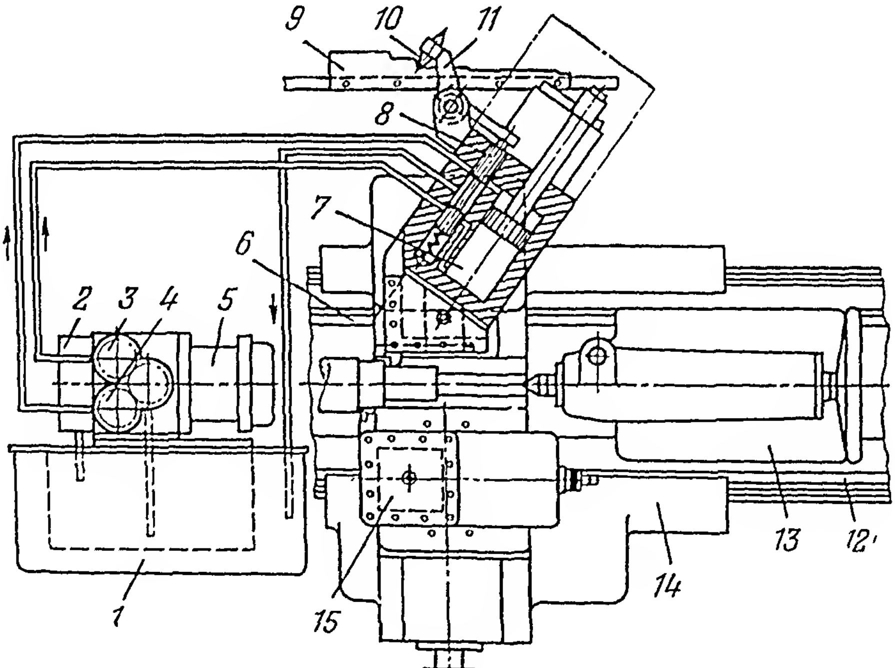


Fig. 135. Hydraulic tracer-controlled slide for rear installation:
1—tank; *2*—filter; *3* and *4*—double gear pump; *5*—electric motor; *6*—tracing slide carriage; *7*—power cylinder; *8*—tracer valve; *9*—template; *10*—stylus; *11*—stylus holder; *12*—lathe bed; *13*—tailstock; *14*—carriage; *15*—square turret

Front installation of the slide (Figs. 133 and 134) is convenient in that the template is mounted in front. On the other hand, the template covers the cutting zone in this case, and there is no standard slide with which a model workpiece can be turned.

In the Soviet Union, hydraulic tracer-controlled slides are available as separate units (attachments) that can be mounted on the lathe saddle. Model ГС-1 is mounted at the front (see Fig. 39 of Vol. 1), while models КСТ-1, ПКТ-3 and КШ are installed at the rear.

If the tracing slide is mounted in front, the lathe spindle rotates in the usual direction. When the slide is installed at the rear the direction of spindle rotation depends upon how the cutting tool is set up and clamped. If the tool is clamped "normally" (with the face upward) reverse spindle rotation is required; if it is clamped upside-down (with the face downward) forward spindle rotation is required.

CHAPTER 13

APPARATUS FOR HYDRAULIC SYSTEMS OF MACHINE TOOLS

13-1. General Data

Hydraulic apparatus of various types and designs are used to control the parameters of the fluid flowing in the hydraulic system, i.e. the pressure and volume of the flow, and also for shutting off certain parts of the circuits from others.

All apparatus used in the hydraulic systems of metal-cutting machine tools can be classified according to its main function as pressure or volume control valves and directional control valves; according to its principle of operation as shutoff, reducing, relief, check and other valves; and according to its construction as plunger, spool, rotary and other types of valves.

A valve is an apparatus (hydraulic device) mounted in the path of fluid flow and designed for changing the parameters of the flow.

Each valve has an operative member which is actuated either by an external force (adjustable valves) or by the fluid passing through it (nonadjustable valves). This action changes the cross-sectional area of the opening in the valve, thereby changing the parameters of the fluid flow.

Pressure losses in valves depend upon oil velocities v and v_v and the resistance coefficients ξ and ξ_v of the inlet port of the valve and the valve opening, respectively. Hence, the pressure loss is equal to the sum

$$\xi \frac{v^2}{2g} + \xi_v \frac{v_v^2}{2g} = \xi_0 \frac{v^2}{2g}$$

where ξ_0 is the total resistance coefficient

$$\xi_0 = \xi + \xi_v \left(\frac{v_v}{v}\right)^2 \tag{205}$$

The velocity v in the inlet port of the valve is usually taken as 7 or 8 m per sec; that in the valve opening from 15 to 25 m per sec, and in relief valves—up to 30 m per sec.

The resistance coefficient of the inlet port of the valve varies in the range $\xi = 2.5$ to 7.

According to the continuity equation

$$v_v F_v = \psi v F$$

where v = velocity in the inlet port of the valve
F = cross sectional area of this port
$\psi = 0.9$ to 1 = factor taking into account the nonuniformity of velocity distribution over the area of the valve opening

Then

$$\frac{v_0}{v} = \psi \frac{F}{F_0}$$

and therefore

$$\xi_0 = \xi + \xi_0 \left(\psi \frac{F}{F_0}\right)^2 \tag{206}$$

Valves whose function requires that they offer least resistance to oil flow (for example check valves) should have an opening with an area $F_c \simeq 0.75 F$. The corresponding velocity ratio is $\frac{v_0}{v} \leq 1.2$ to 1.3

Fluid pressure is controlled by relief, safety relief and reducing pressure control valves

Relief valves maintain a constant pressure at the valve intake by continuously discharging a certain amount of oil back to the tank. Leaktightness is not required of such valves; they are used in all types of flow controls

The purpose of a safety relief valve is to prevent the pressure from building up above a given maximum value. Such valves must be leaktight because they open only upon a pressure rise discharging a portion of surplus oil. Safety relief valves are used in variable displacement speed control circuits

Reducing valves reduce the oil pressure and maintain the reduced pressure at the valve outlet

Due to the effects of friction and leakage a valve cannot react to arbitrarily small changes in flow. This means that each valve has an insensitivity zone

The following factors have a bearing on the degree of insensitivity

(a) a force acting on the side surface of the valve spool and due to the asymmetric flow of fluid through the radial clearances between the spool and the bore in the valve body so that nonuniform and one sided pressure occurs (observations show that this force increases with the fluid pressure)

(b) friction between the moving member of the valve and its guide surfaces

(c) lateral pressure due to not entirely symmetric distribution of the force exerted by the spring

The last factor is dependent on the method of seating the spring. The use of spherical bearing members greatly reduces lateral pressure

The lateral forces due to asymmetric radial clearances can be reduced by providing a series of annular grooves 0.3 to 0.[illegible] mm deep and 0.5 [illegible] mm wide on the cylindrical surface of the plunger [illegible]. The [illegible] relieved of these forces to an even greater extent if a single wide groove 0.[illegible] mm deep is made in the middle part instead of a series [illegible]

When a safety relief valve is closed the force exerted by the spring should exceed the force of the oil pressure on the spool by a certain amount P_{lt} to ensure that valve is leaktight.

Relief valves can be adjusted for a pressure range not exceeding 4 : 1.

Soviet machine tool designers select valves according to ENIMS standards which list the maximum recommended oil flow, working pressure and port size which corresponds to the size of the hole in the mounting connection of the valve. It should not be less than the internal diameter of the pipe to which the valve is connected.

It should be noted that the internal diameter depends upon the pipe wall thickness, the outside diameter being determined by the diameter of the external thread according to the pertinent standards. Therefore, the port size and internal diameter may not coincide in value.

In accordance with the method used to mount hydraulic apparatus in the piping of machine tool hydraulic systems, it is classified as flanged, external screwed or internal screwed.

13-2. Pressure Controls

Pressure controls are designed for limiting the pressure in any part of the system, for unloading a pump when the preset pressure is reached, for building up backpressure in the exhaust line of a reciprocating or rotary hydraulic motor, for bleeding off surplus flow from the pump to maintain a constant pressure in the system, and for reducing the pressure.

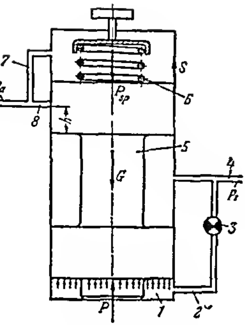
Safety relief valves that protect the system against excessive pressures are not frequently operated. By contrast, bypass, relief, sequence and backpressure valves operate continuously and should therefore have working surfaces made of more wear-resistant materials.

Though there are very many designs of pressure control valves, only two designs have been standardized in Soviet machine tool engineering: with the spool (Fig. 136) and the piston (Fig. 137) types of closing members.

In the spool type of pressure control valve, the pressure is applied through passage *4*. The fluid passes further through damping hole *3* and passage *2* to chamber *1*.

If spool *5* is in equilibrium, then force P_{sp} of spring *6* and the dead weight of the spool counterbalance the force of pressure P and the friction force S. When equilibrium is upset, i.e. when $P > P_{sp} + G + S$, the spool is lifted and connects passage *4* through passage *8* with the tank. Leakage drains back to the tank through passage *7*.

The overlap h (Fig. 136) in such valves is usually equal to $h \cong \frac{d}{3}$, where d is the spool diameter.



ig 136 Spool type valve model Г54

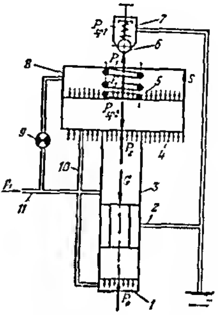


Fig. 137 Piston type valve, model Г52

The spool is lifted after the pressure in the system has been built up to

$$p_{max} = \frac{c(h_0 + h) + P + S}{F} \tag{207}$$

ere c = rigidity of the spring (load corresponding to unit deflection)
h_0 = initial compression of the spring
F = area of the spool directly contacting the fluid under pressure $\left(F = \frac{\pi d^2}{4} \text{ in our case}\right)$

The change in the force exerted by the spring on the spool is dependent upon change in spool travel as follows

$$c(h_{max} - h_{min}) = P_{max} - P_{min}$$

f the closing member is of the piston type (Fig. 137) the fluid flows through sages *10* and *11* to chambers *1* and *4*, and simultaneously through damper o the upper chamber *5* in the valve. As long as force P_1 due to the fluid ssure acting on ball valve *6* is balanced by the force P_{sp1} of valve spring *7*, ton *3* is in a state of equilibrium and

$$P_3 + P_{sp2} + G = P_0 + P_2 \pm S$$

where P_3 = force of the pressure on the upper area of piston *3*
P_{sp2} = force exerted by spring *5*
G = dead weight of piston *3*
P_0 = force of the pressure on the area *1* of piston *3*
P_2 = force of the pressure on the area *4* of piston *3*
S = force of friction.

Since $P_3 = P_2 + P_0$, then $P_{sp2} = \pm S - G$. Thus, spring *5* balances the difference between the friction force S and the weight of piston *3*.

If $P_1 > P_{sp1}$, valve *6* opens, the pressure in chamber *8* drops and the sum of forces $(P_0 + P_2)$ becomes more than force P_3. Under the action of this difference in forces, piston *3* is lifted and passage *11* is connected through passage *2* with the tank to which surplus flow is exhausted.

Damping device *9* is actually built into piston *3*, and its purpose is to provide a difference in pressures at the moment the piston is lifted and, at the same time, to damp its free vibrations (to eliminate chatter).

Such valves are statically balanced (they are called balanced-piston-type relief valves) so that even a small pressure drop immediately leads to the lifting of piston *3* to connect the pressure line to the exhaust passage. These valves are capable of maintaining stable pressure through their full range of adjustment.

Chamber *8* can be connected to a globe valve or any other control device; this enables the pressure in the hydraulic system to be remotely controlled or, if necessary, the system may be unloaded after each cycle of the machine.

Reducing valves are used in hydraulic machine tools (or in other machines) to reduce the pressure in the system, and also when it is necessary to divide the delivery of a pump into several parallel lines, requiring different pressure levels, for instance, for a lubricating system, for controlling directional control valves, clamping devices, etc.

In cases in which the flow at reduced pressure is to be restricted in volume as well, the reducing valve is combined with a throttle device connected in series or in parallel with the chamber of reduced pressure.

Reducing valves may have either of two functions in a hydraulic system: they may maintain a fixed reduced pressure in a part of the circuit, regardless of the pressure level in the rest of the circuit, as for example in a speed stabilizer, or flow-control valve with pressure compensator (Fig. 121), or they can maintain a fixed difference of pressure. In the latter case, the reduced pressure varies with the pressure in the rest of the system.

The closing member of the reducing valve shown in Fig. 138 is of the piston type, resembling the piston of the relief valve in Fig. 137. The damping action is applied at the fluid input in relief valves; in reducing valves the reduced pressure p_2 undergoes damping. This is the output flow and, for this reason, two damping arrangements *9* and *10* are provided in the circuit of Fig. 138.

The required reduced pressure p_{*} is set by adjusting spring *7* (Fig 138) Oil at high pressure p_1 is delivered to port *1* ond passing through the valve opening formed by piston *2* and the valve body, has its pressure reduced to p_{*} From passage *11* fluid at a pressure of p_{*} is discharged to the system and is admitted to chamber *13* through passage *12* and to chamber *3* through damping device *10* Oil also passes through domping device *9* into chamber *8*

As long as ball valve *6* is held closed by the action of spring *4* tho forces of the pressuro abovo and below piston *2* are balanced Upon an increase in pressure p_{*} tho forco P_1 acting on the ball valve increases so that $P_1 > P_{sp1}$ At this tho ball valve

c

F

r

than in chamber *8* As a result piston *2* is lifted closing (or reducing) input port *1* and shutting off the high pressure line from the low pressure one

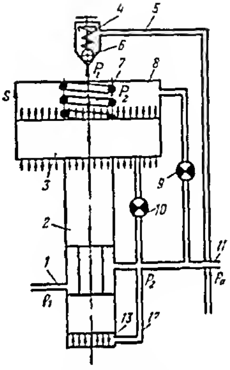


Fig 138 Diagram of reducing valve model Г5

In cases when a power cylinder or rotary hydraulic motor must provide rapid traverse movements special hydraulic apparatus is used This apparatus is either in travel controlled as from a cam or changes in pressure in the system are used for this purpose

Rapid traverse is effected by a cam operated valve by utilizing fluid flowing out of the backpressure chamber or by using an additional pump

In a circuit with in travel controls a two position directional valve with o check valve model Г74 3 (Fig 139) is used At rapid approach of the power piston spool *4* is held in the lifted position by spring *1* and fluid can pass freely out of the cylinder end with backpressure p At this time the backpressure drops to its minimum value and the whole output of the pump is delivered to the head end of the power cylinder When the cam depresses spool *4* port *2* is closed chamber *3* is shut off from the drain line and the fluid forced out of the cylinder must pas only through flow control valve *C* At this a backpressure is built up the pre sure increases in the pressure line part of the fluid drains back to the tank through valve *7* and piston speed is reduced

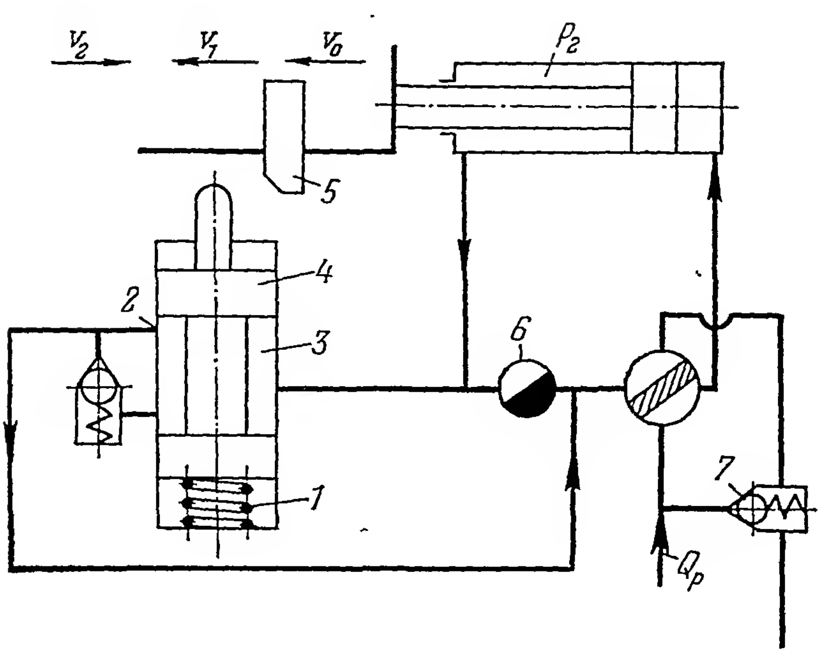


Fig. 139. Circuit with a two-position two-way directional valve, model Г74-3

If the cam is replaced by a template of the proper profile, the speed of the power piston can be changed several times during one stroke.

The velocity increase factor for the power piston is

$$k_0 = \frac{v_0}{v_{1max}} = 1 + \frac{q}{Q}$$

where q is the flow through the relief valve at maximum working speed.

The full output of the pump $Q_p = Q + q$ is utilized only for rapid traverse movements of the power piston; during the working stroke, a part of the fluid is drained back to the tank through the relief valve. As a result, the fluid is more intensively heated and the efficiency of the machine tool is reduced. For these reasons, such a circuit is recommended only for $k_0 \leqslant 1.3$ or 1.4.

If the working cycle of the machine requires a velocity increase factor $k_0 > 4$, an additional pump for rapid traverse is provided besides utilizing the oil exhausted from the cylinder. This combination of two methods reduces the required output of the additional pump at the same velocity v_0 by from 60 to 67 per cent.

The additional pump is cut out automatically at the beginning of working travel by using a control panel of the type shown in Fig. 140 (model Г53-1).

This panel is a combination of a high-pressure relief valve *3*, type Г52 (see Fig. 137), dividing valve *5* and check valve *7*, model Г51. The body of the panel has four connections: inlet *1-4* is connected to the working travel pump—a high-pressure pump with a delivery Q_1, outlets *2* and *6* are connected to the tank, and inlet *8* is connected to the additional pump with a delivery Q_2.

Let us denote the pressure developed by force P_{sp} of the spring in dividing valve *5* by p_{sp}. Then

$$p_{sp}=\frac{4P_{sp}}{\pi(D_1^2-D_2^2)}$$

where D_1 and D_2 = larger and smaller diameters, respectively, of the piston in valve *5*

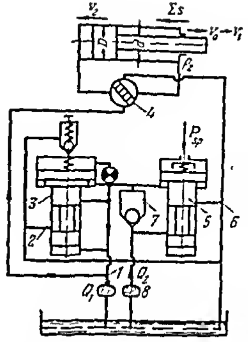


Fig. 140. Circuit with a control panel of the dividing valve type, model Г53

Let us denote the pressure developed in the hydraulic system due to the total force $\sum S$ of all detrimental resistances by p_s. Then

$$p_s=\frac{4\sum S}{\pi d^2}$$

where d is the piston rod diameter

Rapid traverse is possible under the condition that

$$p_{sp}>p_s$$

In this case, check valve *7* opens, the output of the two pumps is combined at point *1* and delivered to the cylinder. When the piston is switched over to working travel, backpressure p_2 is developed. The pressure in the cylinder increases ($p_s > p_{sp}$), and the piston of valve *5* is lifted due to the increased pressure acting on the area $\frac{\pi}{4}(D_1^2-D_2^2)$. This connects the pump with exhaust Q_2 to the tank through line *6*. After this, only oil from the high-pressure pump is delivered to the cylinder.

The velocity increase factor for this circuit is

$$k_0=\frac{v_0}{v_{1max}}=\frac{4(Q_1+Q_2)\pi D^2}{4Q_1\pi d^2}=\frac{D^2}{d^2}\left(1+\frac{Q_2}{Q_1}\right)$$

For $d=(0.5 \text{ to } 0.7)\ D$, i.e. $\left(\frac{D}{d}\right)^2 \cong 2 \text{ to } 4$,

$$k_0=(2 \text{ to } 4)\left(1+\frac{Q_2}{Q_1}\right)$$

To ensure a dependable rapid traverse, it is advisable to take $p_{sp} \cong$ $\cong (1.15 \text{ to } 1.2)\ p_s$. Consequently, the required force of the spring in dividing valve *5* should be

$$p_{sp} \cong (1.15 \text{ to } 1.2)\,\frac{D_1^2-D_2^2}{d^2}\sum S$$

13-3. Directional and Volume Controls

Hydraulic reversing devices for machine tools have important advantages over mechanical or electrical reversing systems, in that they are superior as to speed of response and permissible frequency of reversals. For these reasons, in hydraulic machine tools, such reversing devices have almost completely replaced all other types.

Spool-type directional control valves are extremely simple in construction. They usually consist of a cast-iron housing with the required ports, port slots and passages for fluid flow, and a hardened and ground, and sometimes lapped, spool, with two or three lands for diverting the fluid, thereby connecting the source of supply with the corresponding lines.

In accordance with the number of exhaust ports (there is always one input port), directional valves may be two-way, three-way, four-way, five-way and multiple-path types.

A two-way valve (Fig. 139) is employed in the simplest cases for either blocking or admitting flow from one port to another.

Three-way directional valves (Fig. 141) are employed for alternating the direction of flow to two or several energy consumers. Thus, in rapid forward traverse of the piston, along arrow v_0, the three-way valves *3* and *4* (model Г73-2) are switched off (the solenoids are de-energized). Oil can be freely discharged from the cylinder. When flow-control valves *1* and *2* are cut in simultaneously or separately, the speed of the piston is reduced to $v_{1.2}$, v_2 and v_1, respectively.

On the return stroke of the piston (after reversal), the oil passes through the exhaust ports of the directional control valves to the right (rod) end of the cylinder.

Four- and five-way valves always have two ports that can be connected alternately either to the pressure or to the exhaust line. Central port *1* (Figs. 142 and 143) is connected to the pressure line from the pum... ports *2* and *3* are connected to the input and exhaust of the hydraulic ...

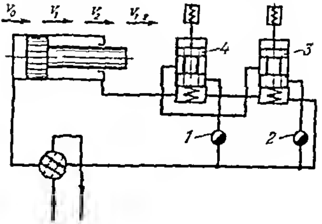


Fig 141 Circuit with two-position three-way directional valves model 2Г73 2

Ports 4 and 5 are connected to the tank In a four-way valve, ports 4 and 5 are connected together and to a common exhaust line The two ports are separated in a five-way valve

If, in entering the valve through port 1 (Fig 142), the oil flow does not meet a land of the valve spool, then the direction of spool movement (for instance, to the left) coincides with flow from the pump On the other hand, if the spool land meets oil flow (Fig 143), then flow and spool movement do not coincide In this case, the valve will be somewhat less simple and require more space

Valve spools are made of low-carbon steel which is properly carburized and hardened Lapping is rarely resorted to, the spools are usually ground to a tolerance in the range from 0 004 to 0 013 mm The diameter of the

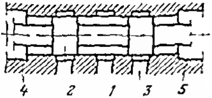


Fig 142 Four (five) way two (three) position directional valve in which directions of fluid flow and valve spool movement coincide

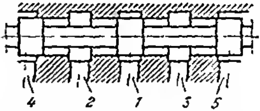


Fig 143 Four (five) way two (three) position directional valve in which directions of fluid flow and spool movement do not coincide

For $d=(0.5 \text{ to } 0.7)\ D$, i.e. $\left(\frac{D}{d}\right)^2 \cong 2 \text{ to } 4$,

$$k_0=(2 \text{ to } 4)\left(1+\frac{Q_2}{Q_1}\right)$$

To ensure a dependable rapid traverse, it is advisable to take $p_{sp} \cong \cong (1.15 \text{ to } 1.2)\ p_s$. Consequently, the required force of the spring in dividing valve *5* should be

$$p_{sp} \cong (1.15 \text{ to } 1.2)\ \frac{D_1^2-D_2^2}{d^2} \sum S$$

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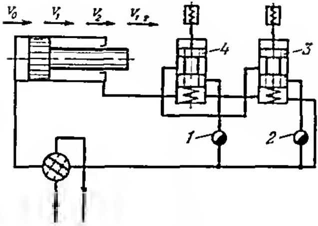

Fig. 141. Circuit with two position three way directional valves model 2ЬГ73 2

Ports *4* and *5* are connected to the tank. In a four way valve, ports *4* and *5* are connected together and to a common exhaust line. The two ports are separated in a five way valve.

If, in entering the valve through port *1* (Fig. 142), the oil flow does not meet a land of the valve spool, then the direction of spool movement (for instance, to the left) coincides with flow from the pump. On the other hand, if the spool land meets oil flow (Fig. 143), then flow and spool movement do not coincide. In this case, the valve will be somewhat less simple and require more space.

Valve spools are made of low carbon steel which is properly carburized and hardened. Lapping is rarely resorted to, the spools are usually ground to a tolerance in the range from 0.004 to 0.013 mm. The diameter of the

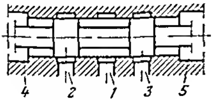

Fig. 142. Four (five) way two (three) position directional valve in which directions of fluid flow and valve spool movement coincide

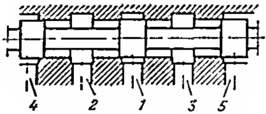

Fig. 143. Four (five) way two (three) position directional valve in which directions of fluid flow and spool movement do not coincide

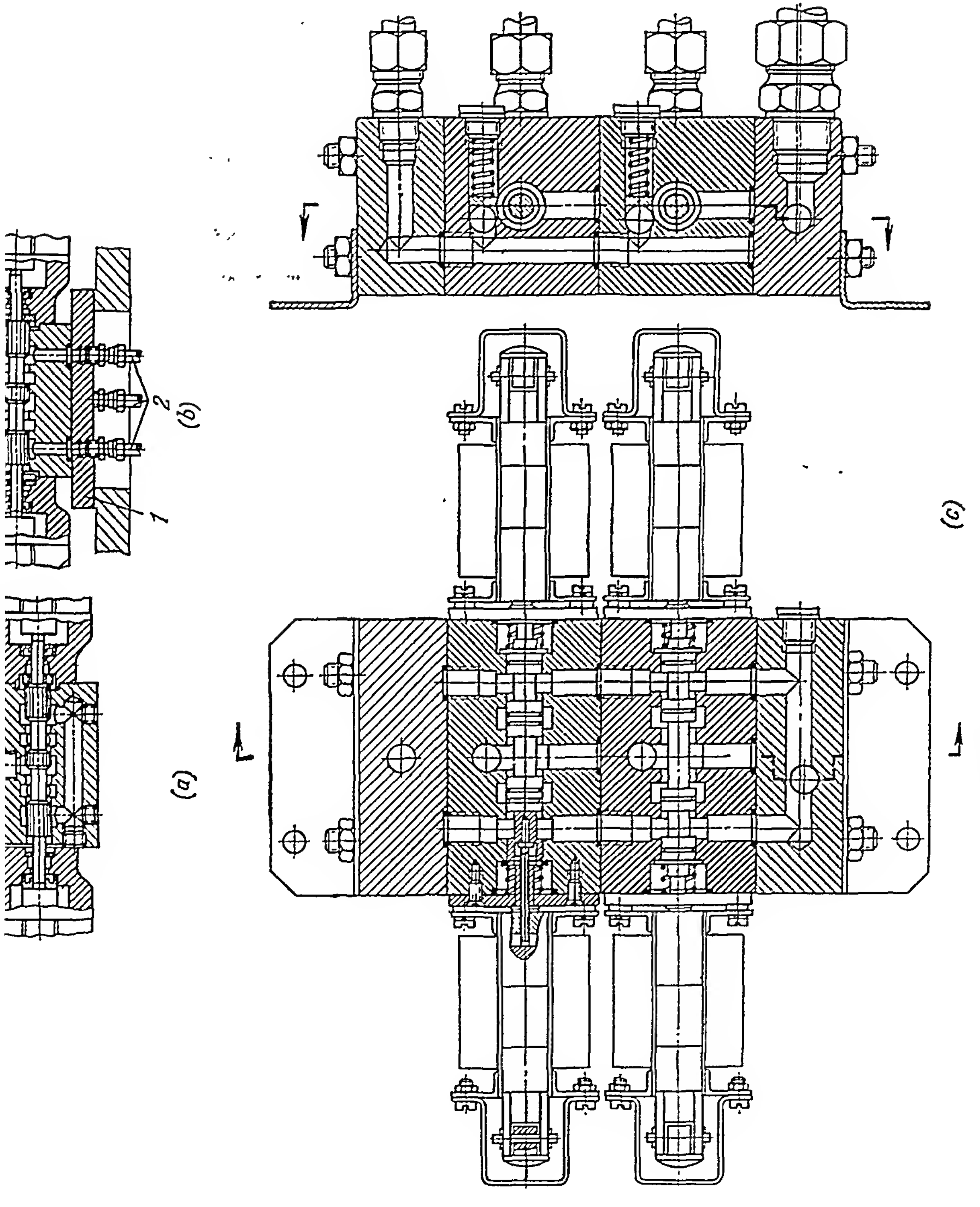

Fig. 144. Examples of valves with ordinary connections for piping (a) and gasket-mounted valves (b and c).

internal passages and ports in the housing should be $d = (0.75$ to $0.85)\ d_{pp}$, where d_{pp} is the size of the piping.

Recently gasket-mounted valves (with pipeless connections) have gained in popularity (Fig. 144b), as a modification of the basic model (Fig. 144a) for ordinary connection to piping or tubes. In the gasket-mounted design, all the port connections are brought out to an accurately machined surface, counterbored for O-ring gaskets, which mates with subplate *1*. Pipe connections *2* at the rear of the subplate coincide with the valve ports and connect the valve to the rest of the hydraulic system.

The arrangement and type of directional control valves are selected to suit the operating cycle of the machine tool, the accepted sequence of machining operations and the method to be used for controlling the cycle (for instance, manual, electrical or hydraulic controls). From this aspect, all directional control valves are classified as two-, three- and multiple-position valves with pilot (hydraulic or solenoid), manual or solenoid control.

Two-position valves are commonly employed in machining work in several passes, for example, in grinding machines. The spool in such valves has only two positions, it may be shifted either to the extreme right or extreme left (Fig. 145).

Three-position valves find application in machining work in a single pass (drilling, milling, etc.), as well as in various clamping and handling devices and, in general, when a short interval is required, for example, to remove the finished work and to load and clamp the next blanks, for replacing tools, etc. (Fig. 146).

Multiple-position valves (Fig. 147) are frequently used in machine tools in which the command signals change automatically. Examples are multiple-spindle unit-built machines that operate in a single pass. Common command signals in such cases are: RAPID APPROACH, one or two WORKING FEEDS, RAPID WITHDRAWAL and STOP.

Two-, three- and multiple-position valves may be hydraulic pilot-operated (Fig. 145), solenoid-operated (Fig. 146) or hand-operated type (Fig. 148).

In a pilot-operated two-position valve model Г72 (see Fig. 145), spool *1* is shifted by pressure from the main line which, however, should not exceed 100 kg per sq cm. At higher pressures, a reducing valve is installed into the control line. Sometimes, a small displacement low-pressure pump is provided for this purpose.

Ball-type check valves *2* and *11* are mounted in the end covers of the housing with pin chokes *3* and *10*. These arrangements adjust the speed of valve spool *1* independently in each direction, varying the valve shifting time in a range from 0.3 to 3 sec. Such adjustment permits pilot-controlled reversal to be applied at any speeds of the hydraulic motor.

Applying pilot control to directional control valves enables two speeds of spool travel to be obtained (model 4Г72, Fig. 145a).

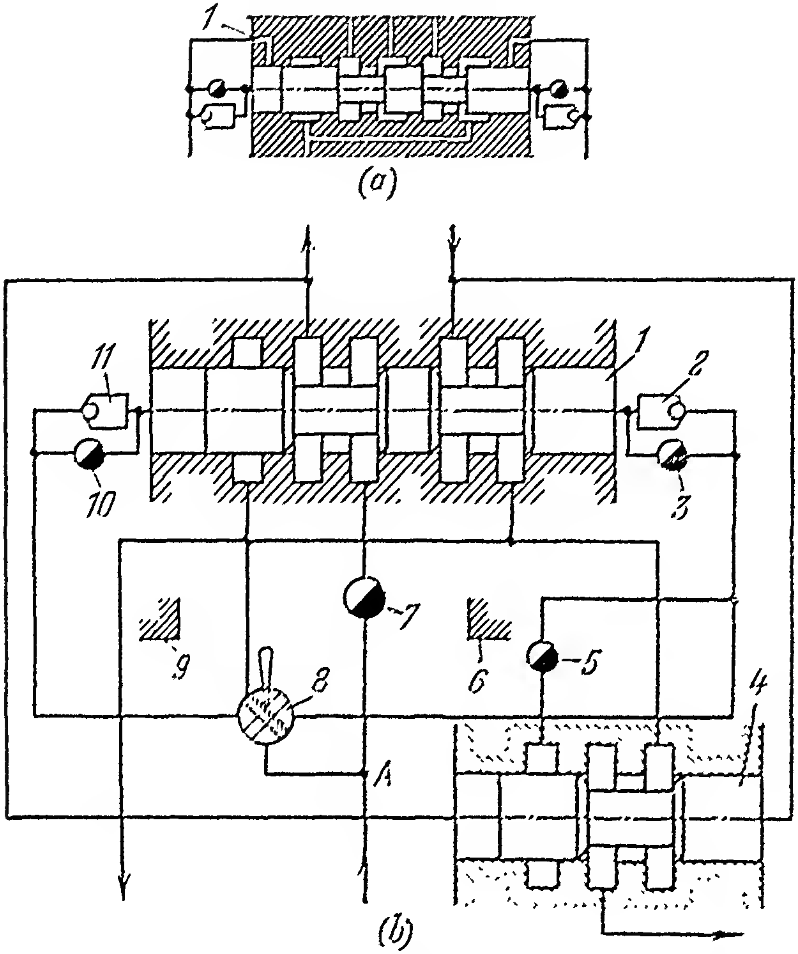

Fig. 145. Control circuit with a two-position four-way directional valve, model Г72-1 or 5Г73

In the initial moment, the spool shifts rapidly since oil is forced out through auxiliary channel *1* in the valve housing. When this channel has been blocked off by the spool, the speed of the spool is reduced.

Experiments have shown that this change in the speed of valve motion can substantially reduce overtravel of the workpiece at the moment the hydraulic motor (reciprocating or rotary) is reversed.

The command signal for reversal is initiated by rotary pilot valve *8* (model Г71-21), which is actuated by cams *6* and *9* of the table.

The reversing line should always be independent of the motor speed control line. This can be achieved by connecting the valves in parallel, selecting

the branching point A before the flow control valve 7 and thus obtaining two entirely independent oil flows

To obtain intermittent (increment) feeds (as for instance of the wheel in surface *grinding milling cutter in die sinking etc*) portioning valve *4* is provided in the circuit. This valve periodically admits oil to the feed cylinder. The piston of this cylinder travels at a speed that depends upon the oil flow through flow control valve *5*

Two self centring pistons 1 (Fig 149) are provided in three position directional *control valves. These pistons have the form* of sleeves sliding in the bore of the valve *housing and the valve spool can slide in the* sleeves. The spool is located in its third or mid position when oil under pressure is admitted to the left and right end *2* and *4* of the valve simultaneously

If pressure is applied only to one end for example in *2* and end *4* is connected to the *tank then the self-centring piston 1* and spool *3* begin to shift simultaneously to the right until the flange of the piston reaches the counterbore in the housing. The spool travels further alone

Hand operated directional valve model 174 *(see Fig 148) have a positive location feature for the spool by means of a spring* loaded ball detent *1*. They register two position valves in the two extreme positions and three position valves in the required three positions. In one modification of this valve centring springs return the spool to the central position from either of the extreme (hard over) positions as soon as the control lever is released. In such cases a detent is combined with a self-centring piston

Regardless of the mode of operation each type of three position valve has several modifications based upon the width b of the land on the spool width b_1 of the port opening in the housing and the ratio of the lengths B and B_1 (Fig 146). These modifications are

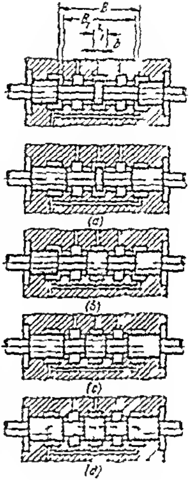

Fig. 149. Diagrams of three position four way directional valves models 1Г73, 2Г73, 3Г73, [illegible]Г73 and 6Г73

time is possible so that shocks may occur if hydraulic motors are revers at high speeds.

In a multiple-position valve (Fig. 147), the spool has a stepped shank, b, c, d and e, the number of steps corresponding to the number of comman signals required to control the hydraulic motor. The direction and spe of the motor (rotary hydraulic motor or cylinder) is determined by th position of the main valve spool *1*. The latter is shifted in either directio by an oil pressure of $p = 20$ to 40 kgf per sq cm applied in port *4* of th pilot valve *2*. Pilot valve spool *2* is shifted to the left by a solenoid and t the right by spring *3*. During the working cycle of the machine tool, the main valve is tripped by cams *6* and *8*. As roller *5* of lever *7* runs up on the cams, lever *7* turns lever *9* whose end enters a slot of detent *10*. The detent is lifted, and the force P of fluid pressure shifts the valve spool until the next step runs up against the detent. At the last step, the hydraulic motor is stopped.

The power piston is switched over to forward rapid traverse by energizing the solenoid which shifts the pilot valve spool to the right. At this, fluid at a pressure of $p = 20$ to 40 kgf per sq cm is admitted at the stepped end of spool *1* which is shifted to the extreme right position. Then the power piston travels forward rapidly until it reaches working feed cam *6*. This is repeated each time roller *5* of lever *7* runs up on a cam and lifts detent *10* the height h of the corresponding step.

First position *a* of the main valve spool corresponds to rapid forward traverse of the hydraulic motor; the second *b* and third *c* positions are the first and second working feeds. The fourth position *d* corresponds to rapid return of the piston and the fifth position *e* is the STOP command signal.

13-4. Reversal of Reciprocating and Rotary Hydraulic Motors

The construction of devices for reversing the movements of operative units is of prime importance in machine tools with a reciprocating main motion. High table speeds—up to 60 m per min in up-to-date surface grinders and up to 80 m per min, or even more, in planers—in conjunction with the motion of large masses and the substantial cutting forces, are the causes of the complex effects that occur in the reversal period. The use of an improperly designed reversing device may lead to shocks, nonuniform travel of the operative units (for instance, table), and excessive overtravel values.

Reversing arrangements, in which variable-displacement pumps are used in a combination with the hydraulic motor, constitute a closed system. Reversing is accomplished by changing the direction of oil flow. When lever *1* is swung over (Fig. 150), the retaining ring or slide block is shifted through

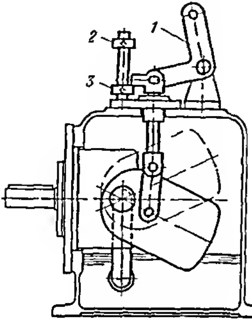

Fig. 150. Arrangement for reversing the fluid pressure in axial piston pumps

its neutral position to the opposite side. As a result, the suction and discharge sides of the pump are interchanged.

Stops *2* and *3* are adjusted to vary the pump output, i.e. the speed of the operative unit, for instance, the table of a planer. This type of arrangement is suitable for machine tools with long strokes and large reversing masses. The kinetic energy of the masses being reversed can be utilized in the pump which operates as a rotary hydraulic motor during table reversals. This feature reduces the heating of the oil and losses of energy.

Reversing valves are used, however, on low power machine tools with relatively short strokes, since a large number of table reversals per minute is detrimental to the operation of the pump which will soon get out of order.

In the Soviet Union, the standard hydraulic control panel, model 131 (Fig. 151), is extensively used in grinding machines and conveying devices. It is based on valve controls and simultaneous braking. The flow capacity of these panels ranges from 18 to 140 litres per min and depends upon the type and size. The working pressure ranges from 5 to 25 kgf per sq cm, pressure loss in passing through the channels of the panel does not exceed 5 kgf per sq cm. The pressure drop in valve control does not exceed 2.5 kgf per sq cm.

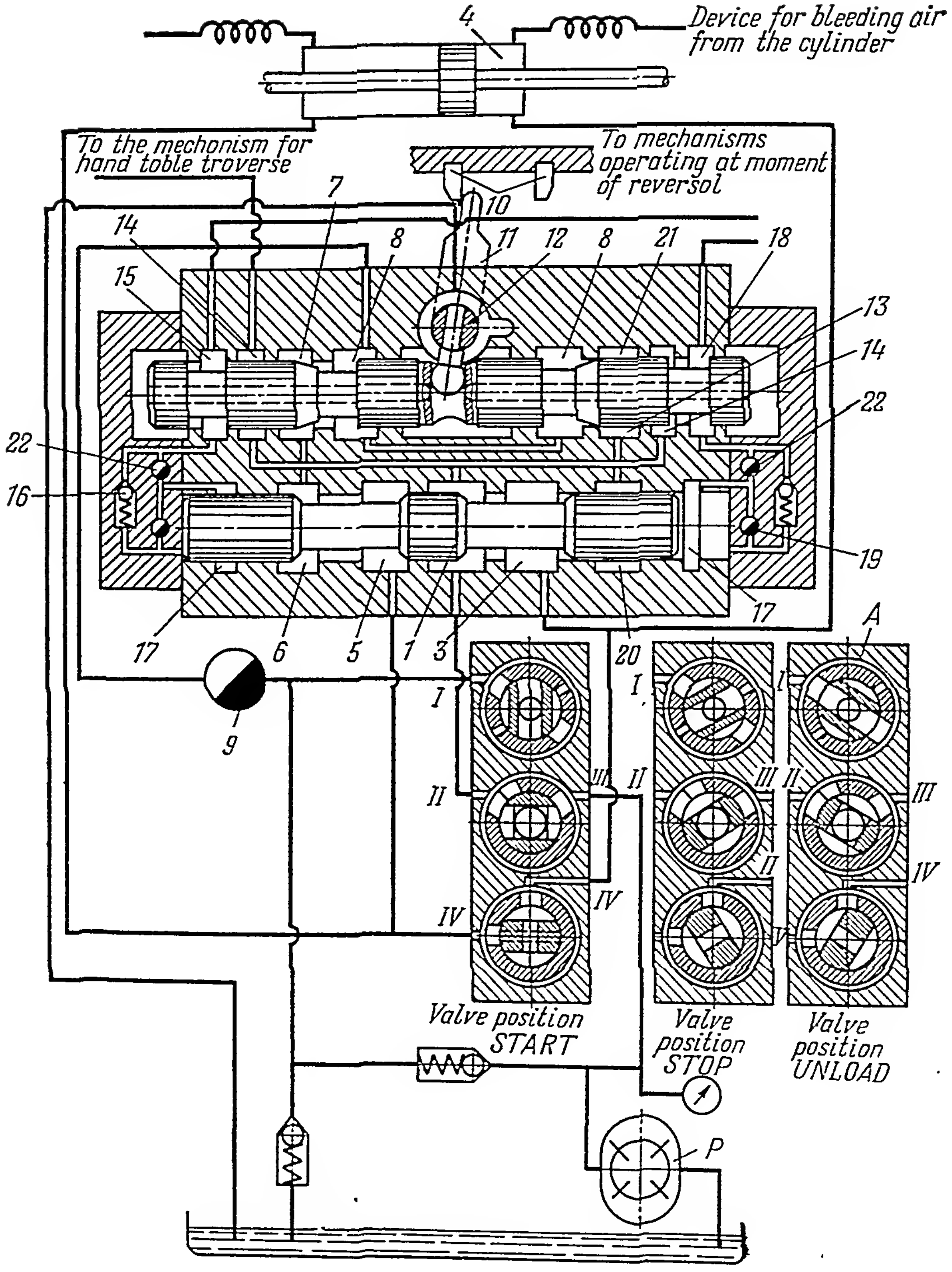

Fig. 151. Diagram of the standard hydraulic control panel, model Г31, for grinding machines

Mounted in the panel housing are valve *13* for directional control and for braking the table directional control valve *1* flow control valve *9* for setting table speeds and rotary valve *A* for START STOP and UNLOAD The side covers contain ball type check valves *16* choke checks *22* for adjusting the braking time and choke checks *19* for adjusting acceleration Valve spool *13* is shifted by means of shaft *12* which is actuated in turn through lever *11*

Oil from pump *P* passes through the rotary valve in the START position and is admitted to port *3* through which it passes to the right end of power cylinder *4* Oil forced out of the left end of the cylinder passes through port slots *5* *6* and *7* and then through port *8* and flow control valve *9* to the tank The power piston travels to the left Cam *10* of the table actuates lever *11* of the table reversing shaft shifting valve spool *13* to the right At this oil from port *14* passes through port *15* and check valve *16* to the left end of valve spool *1* which is shifted to the right During the first part of its path spool *1* travels faster since oil can escape through choke check *22* (on the right side) After the spool has closed port slot *17* its speed is reduced since the oil must then pass consecutively through two choke checks *19* and *22* At the same time the left end of the power cylinder is connected to the pressure line through port *5* while oil forced out of the right end passes through ports *3* *18* *20* *21* and *8* to flow control valve *9* and further to the tank

The reversal of the hydraulic motor i e the shaft of a rotary hydraulic motor or the power piston must satisfy certain definite requirements which can be reduced to the following

(a) the valve spool shifting time should not be dependent on the velocity of the mass being reversed or its kinetic energy

(b) there should be no jerks shocks or vibration in the reversal period

(c) at constant steady state velocity of the hydraulic motor its acceleration and braking paths should be maintained constant in the course of time

(d) provisions should be made for regulating the reversal time in a range of at least 10 1

Inertia forces developed at the moment of reversal may cause considerable hydraulic shocks at high operating speed if the directional control valves have not been properly designed The magnitude of overtravel of the hydraulic motor at reversal depends upon its speed the mass of the parts being reversed and the friction forces in the ways packings etc The most effective means of reducing overtravel is contouring the valve spool to gradually increase the hydraulic resistance of the reversing valve It is necessary to note however that excessive reduction of hydraulic motor speed before the reversal leads to a decrease in the output of the machine tool

During the reversal period the output port from the hydraulic motor is gradually closed by the directional control valve spool If the amount of fluid flowing to this valve port is equal to the amount passing through this

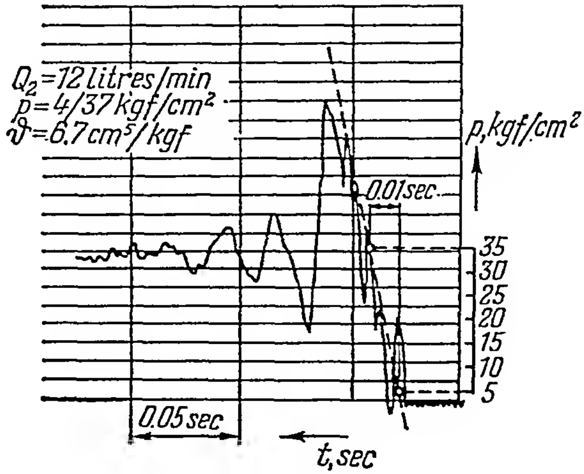

Fig. 152. Backpressure variation during the reversal of a machine tool table

port, the hydraulic motor will be braked only by the friction forces in the motor itself and by the resistance of the valve.

A rise in backpressure and its increase with time are dependent on the elastic constant of the fluid and, in some cases, on the elasticity of the piping.

If Q_2 is the rate of flow to a port of the valve, and Q_3 is the rate of flow through the valve port, then

$$\Delta Q = Q_2 - Q_3 = \vartheta \frac{dp}{dt}$$

where $\vartheta = \vartheta_1 + \vartheta_2 =$ sum of the elastic constants of the fluid and piping (see p. 226).

If the piping is blocked off, $Q_3 = 0$, therefore

$$Q_2 = \vartheta \frac{dp}{dt}$$

from which

$$p = \frac{1}{\vartheta} Q_2 t \tag{208}$$

Tests at almost instantaneous closing of the piping (closing time $\tau = 0.005$ sec) showed (Fig. 152) that the pressure before the valve is not built up instantaneously, but during a certain interval of time. Thus, at a flow of $Q_2 = 12$ litres per min, it takes about 0.01 sec to build the pressure up from 5 to 35 kgf per sq cm.

In addition to the elastic constants of the oil and piping, the time required to built up the pressure is affected by other elasticities (pressure gauge, valves,

etc) and the volumetric losses in the hydraulic system Thus, in the same tests, the elastic constant of the oil was $\vartheta_1 = 4.7$ cm³ per kgf and that of the piping was $\vartheta_2 = 0.5$ cm³ per kgf A direct measurement of the elastic constant of the system gave $\vartheta = 6.7$ cm³ per kgf Hence, the difference between the measured and calculated values was $\Delta\vartheta = 1.5$ cm³ per kgf, or about 22 per cent

The compressibility factor of the oil increases somewhat in the course of operation due to its increase in temperature For example, if the oil temperature increases from 30° to 60°C, its compressibility factor increases from $\beta = 0.75$ to 0.85×10^{-4} cm² per kgf.

The operation of a hydraulic drive of the positive displacement type is based on static principles, i e on the variation of pressure but not on the variation of the amount of fluid, as in a hydrodynamic drive Nevertheless, certain dynamic factors must also be taken into account at high speeds of hydraulic motors

The piping of hydraulic systems in machine tools has, in general, only a small capacity and is of short length, in any case, less than 1 000 m There are no reasons that could lead to variations in the rate of fluid flow in the piping This enables the elastic effect of the hydraulic system to be disregarded, as a first approximation

Making use of equations (140) and (141), and taking the coefficient of resistance of the valve opening area as $\xi = 2$ to 2.5, we can write the continuity equation for the given case Then the braking force is

$$B = F^3\gamma \frac{v_x^2}{2gf_x^2}(1+\xi)$$

where f_x is the current instantaneous area of the directional valve opening or, as in model Г31 grinder control panels (see Fig 151), of braking valve *13*

We can further assume that the velocity v_x of the power piston during the braking period adheres to the following law

$$v_x^2 = v_1^2\left(1 - \frac{x}{H}\right)$$

where H is the travel of the piston during the braking period, and that deceleration of the piston is constant, so that $a = \frac{v_1^2}{2H}$ After substituting these values we have

$$\gamma \frac{F^3}{2g} \frac{v_1^2}{f_x^2}(1+\xi)\left(1 - \frac{x}{H}\right) = M_0 \frac{v_1^2}{2H} + \sum S$$

from which the current area of the valve opening is

$$f_x = Fv_1 \sqrt{\frac{\gamma F(H-x)(1+\xi)}{g\left(M_0 v_1^2 + 2H\sum S\right)}} \qquad (209)$$

If the friction losses in the hydraulic motor are not too high, it can be assumed that

$$2H \sum S \ll M_0 v_1^2$$

and then the approximate solution is

$$f_x = F \sqrt{\gamma \frac{F}{M_0 g} (H-x)(1+\xi)} \tag{210}$$

One of the devices that converts mechanical control signals into electric voltage is the potentiometer, which is applied, for example, in the circuit in Fig. 130 for remote control of the output of a radial-piston pump.

The input member (control) is linked to the movable contact of the reference potentiometer Pot_1 (Fig. 153*a*). The contact of the pick-up potentiometer Pot_2 is linked through a drive with the shaft of the controlling electric motor *2* and through screw *d* to the mechanism for varying pump displacement.

Potentiometers Pot_1 and Pot_2 are connected to the same supply circuit. At the same angular positions of the movable contacts, the voltage between them equals zero. When the input member is turned in either direction, the resulting voltage is proportional to the angle of mismatch of the movable contacts of the potentiometers. Through sensitive relay *F*, this voltage is applied to contactors S_1 and S_2 so that electric motor *2* begins to follow up the reference motion, rotating the contact of potentiometer Pot_2 at the same time.

The polarity of the voltage is determined by the direction in which the input member is rotated.

Potentiometers Pot_1 and Pot_2 should have linear characteristics, thereby ensuring strict proportionality between the angle of turn and the voltage. The characteristic should also be steep, i. e. with a large signal voltage per unit of mismatch.

If the system is supplied from a-c mains, potential regulators are used in place of potentiometers Pot_1 and Pot_2.

In two-stage controls, the signal for displacing the power piston (Fig. 153*b*) is fed in by reference potentiometer *8* and is transmitted by amplifier *9* to electromagnetic transducer *10* which shifts core *11* and servo valve *14* of the first stage of control.

Directional valve *15* serves as the second stage of controls. At point *a*, the pressure of pump *17* is divided into a high-pressure branch for displacing the power piston of the machine, and a branch providing pilot pressure for controlling valve *15*. The pressure is lower in the second branch, since the oil passes through reducing valve *16*.

If servo valve spool *14* shifts to the right, the right end of valve spool *15* is connected to the tank through the lower longitudinal channel in the housing.

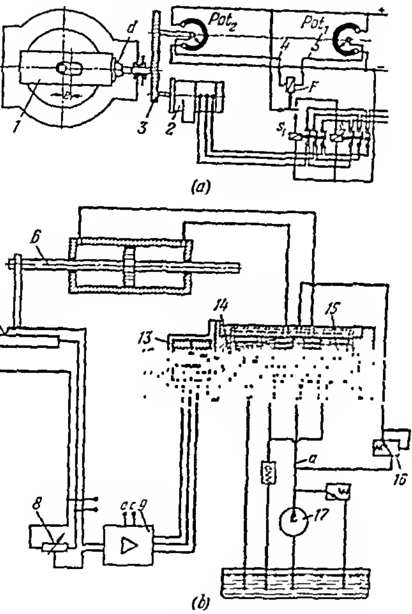

Fig. 153. Control circuits

(a) electro-mech[illegible] by means of a potentiometer bridge circuit [illegible] gearing link of output shaft of the pump [illegible] ecting the two potentiometers; Pot_1—reference [illegible] eccentricity; (b) in-travel control [illegible] feedback potentiometer; 8—reference poten- [illegible] 11—transducer [illegible] 12—control [illegible] valve, 14—reducing valve, 17—pump

The left end of spool *15* is subject to the reduced pressure as before and the spool shifts to the right to admit high-pressure oil to the right end of the power cylinder. At this the piston travels to the left. Feedback from potentiometer *7* ensures the given displacement of the power piston.

13-5. Hydraulic Gear-Shifting Facilities

Gears in gearboxes are shifted by hydraulic means to speed up the shifting process, for remote controls, and when a large force must be exerted for shifting. Hydraulic devices of this type may be a control plunger that is returned by a spring (Fig. 154*a*). In this case, the extreme positions of the plunger are fixed by stops and the sliding cluster gear is held in the left position (in the diagram) by fluid pressure. The last feature is the reason why this construction is almost always used when the gears are to be engaged only for a short time.

In the two-position gear-shifting device (Fig. 154*b*), the plungers are shifted in either direction by oil pressure, up against the housing cover, and hold the cluster gears engaged in these positions.

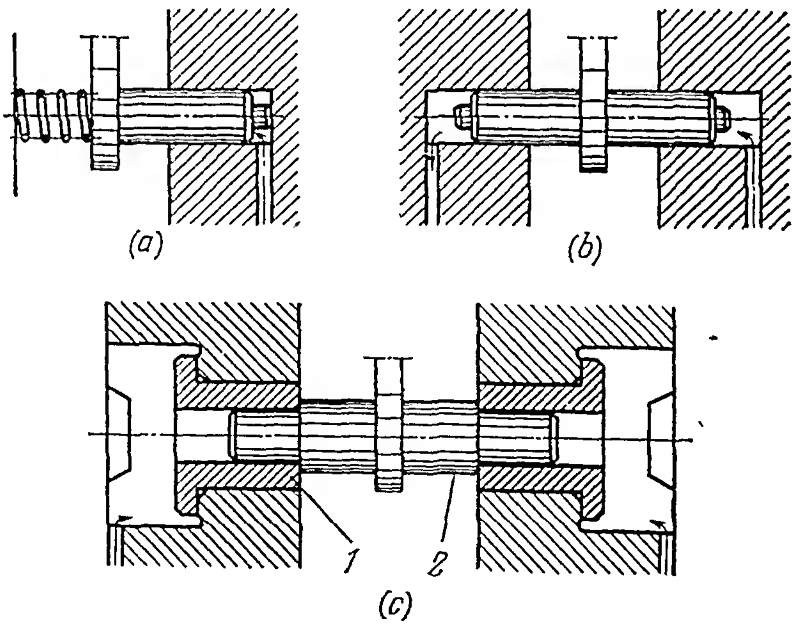

Fig. 154. Hydraulic gear-shifting arrangements for speed gearboxes:
(*a*) with return spring; (*b*) two-position shifting device; (*c*) three-position shifting device

Plunger *2* in the three-position shifting device (Fig. 154c) is linked through a fork with a cluster gear. This plunger can slide in one sleeve *1* and move the other sleeve. The central position of the plunger and cluster gear is obtained by admitting oil under pressure to both the right and left ends of the plunger simultaneously. If pressure is applied only in one chamber and the other is connected to the tank, at first both the sleeve and plunger will be shifted forward until the flange of the sleeve runs up against the counterbore; the plunger will travel further alone. The sleeve diameter is $D = 30$ or 32 mm, the plunger diameter is $d = 25$ mm. The ratios $\frac{a_1}{D}$ and $\frac{a}{d}$ are almost always 2 to 2.5, where a_1 and a are the lengths of the sleeve and plunger, respectively. A pressure of 4 to 6 kgf per sq cm is required to shift cluster gears.

A special rotary valve is used to control oil flow to the plunger cylinders. When the spools of this valve are turned to the corresponding angles, one end of the cylinder is always connected to the pressure line and the other to the tank.

13-6. Master Control Switches and Automatic Operators

The following types of remote controls are extensively employed for controlling semiautomatic and automatic operating cycles of hydraulic machine tools

1. Centralized remote controls, in which the command signals are transmitted to the hydraulic motor as a function of time, so that each subsequent manufacturing operation begins independently of whether the preceding operation has been finished

2. In-travel remote controls, in which each consecutive command signal is transmitted only after the preceding operation has been completed.

Circuits with master control switches usually employ directional control valves of the solenoid operated type (for instance, model Г73-4) having a solenoid of the required pulling capacity, a spool travel of 15 mm and a shifting time $t \cong 0.05$ sec

If in-travel controls are used, the master control switches are designed for sequence-pulse operation Solenoid operated valves *1A*, *1B*, *2A* and *2B* (Fig. 155) are controlled by limit switches 0, 1, 2, 3, 4 and 5. The cycle of machining operations is begun by pressing the START push button either by hand or automatically At the end of each machining operation, a command is transmitted to energize solenoid Sd_1 of valve *I'*. This shifts the spool of valve *I'*, applying the pump pressure to the left end of cylinder *5'*.

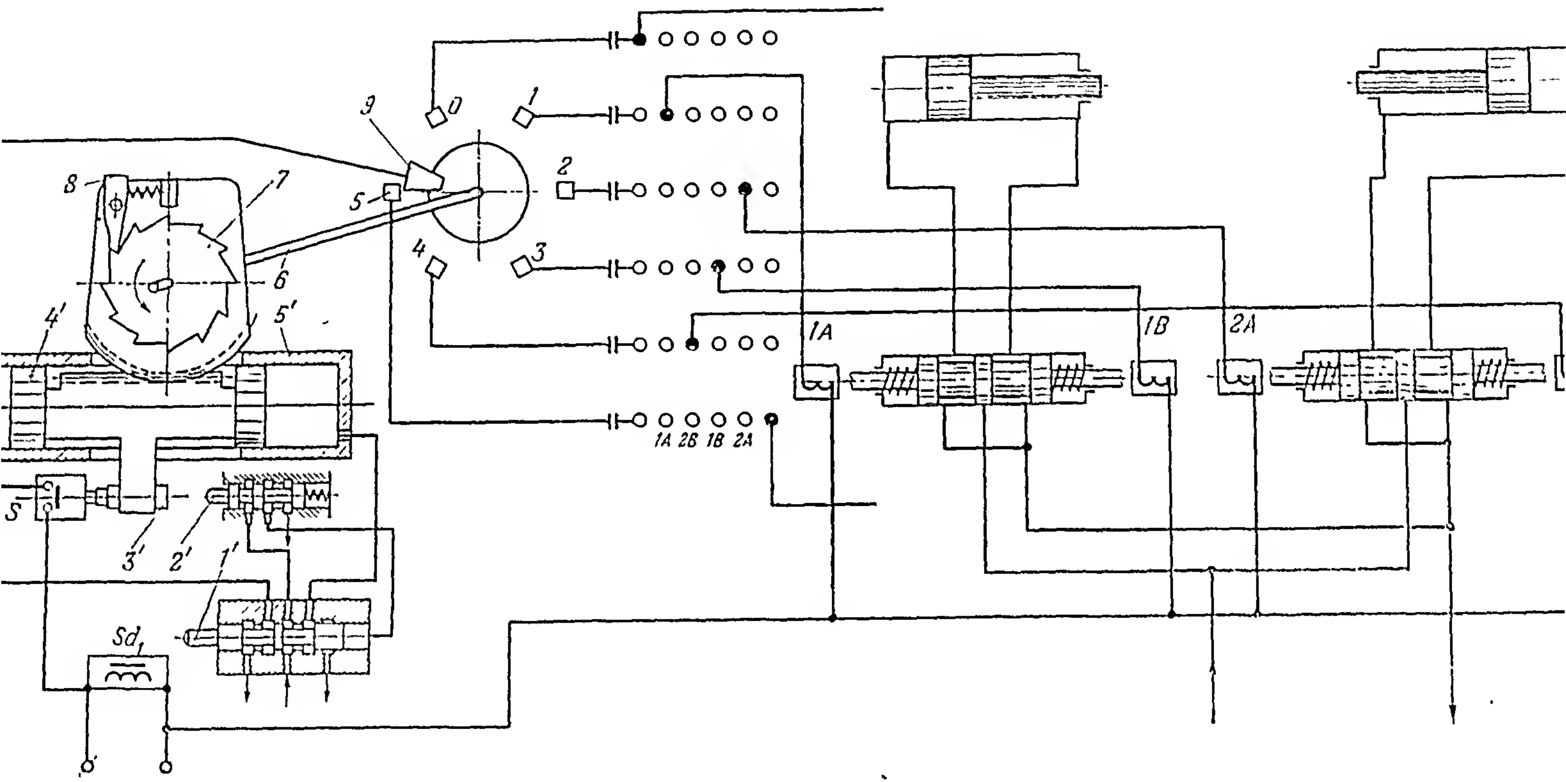

Fig. 155. Diagram of an electrically operated master control switch

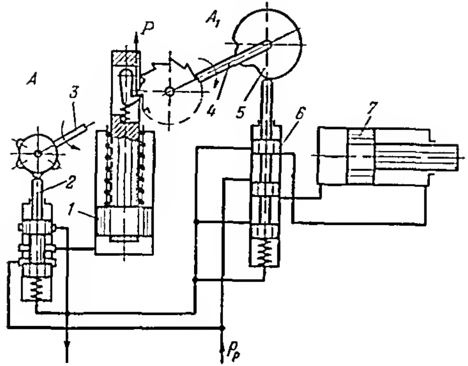

Fig 156 Automatic operator for a lathe type machine tool

Plunger *4'* is shifted to the right and its rack turns ratchet wheel *7*, shaft *6* and sliding contact *9* through an angle corresponding to the next command signal. At the end of the stroke of plunger *4'*, dog *3* actuates return-stroke valve *2*. Oil under pressure is now admitted to the right end of valve spool *1'*, returning it and plunger *4* to their initial positions. This closes the contact of switch *LS* and the master control switch is ready to receive the next command signal. During travel of plunger *4* to the left, shaft *6* of ratchet wheel *7* is stationary and pawl *8* slips over to the next tooth of the ratchet wheel.

One modification of master control switches is the automatic operator (see, for example, Fig 156) which has found wide application in the automation of various loading and handling operations in engine lathes, turret lathes, semiautomatics, as well as in automatic transfer machines.

In most cases the availability of an automatic operator enables a machine tool to be built into an automatic transfer machine or production line. The automatic operator should be located outside of the working zone of the machine tool. This permits the work to be machined using both longitudinal

and cross slides, and the machine tool to be quickly changed over for machining work of other shape and size.

Figure 156 shows a combined diagram of automatic operator *A* and master control switch A_1 as used for sequence control of manufacturing operations. The number of hydraulic motors *7* (power cylinders, in this case), cams *5* and directional control valves *6* should be equal to the number of manufacturing operations handled by the automatic operator.

Four-way directional control valve *2* (model. Г74-2) is used to control hydraulic motors *7*. This valve receives its command signals (is actuated in this case) from cam shaft *3* of the machine tool, or from any other control unit. The pressure p_p developed by the pump is applied in parallel to pusher-piston *1* and to directional control valve *6* of the master control switch. In its upward travel, piston *1* turns cam shaft *4* of the master control switch through a definite angle. Cams *5*, mounted at definite angles in respect to each other, depending upon the cycle of operations, operate valves *6* in the given sequence, thus admitting oil under pressure to the corresponding ends of the hydraulic motors. The motors actuate the required movements in the machine tool.

Investigations and experiments on automatic operators have shown that the level of pump pressure substantially affects the stability with which the hydraulic motors are operated. In this connection, it is advisable to maintain a pressure of p_p = 25 to 30 kgf per sq cm. A decrease in pressure to 18 or 20 kgf per sq cm leads to a stability almost one half that at 30 kgf per sq cm. A rigidity of k = 4 or 5 kg per mm is sufficient for the spring of pusher-piston *1* (Fig. 156); its initial compression should be 3.5 to 4 kg.

The speeds of the hydraulic motors (pistons in power cylinders) in an automatic operator do not usually exceed 120 mm per min.

CHAPTER 14
HYDRAULIC CIRCUIT DESIGN

14-1. Conventional Designation of Pumps, Hydraulic Motors and Other Apparatus in Circuit Diagrams

In drawing hydraulic circuit diagrams for machine tools (and all other machinery), conventional designations are used to represent pumps, reciprocating and rotary hydraulic motors, and all other hydraulic apparatus required to provide the operating cycles of the machine tool. These designations may be of two types.

1. Conventional representation in the form of hydraulic diagrams of each unit of apparatus, in which the principles are shown in some detail, can be used for circuit design and calculations since the method of operation is quite clear. This type of diagram is recommended by ENIMS and is widely used in Soviet machine tool design. One drawback, however, is that much time is required to draw out these diagrams. One simple solution is to draw them out on a separate sheet of paper and to make a large number of copies by blue-printing or other method. Then, in designing a circuit, symbols of the required apparatus can be cut out for pinning or pasting on the diagram, drawing lines representing piping to connect the apparatus in parallel or in series.

2. Graphical symbols, *showing only the functions of each apparatus*, but not its principle of operation, are also used for circuit diagrams. These diagrams cannot be used for design calculations, to follow out the sequence of action, etc. An advantage of these symbols is their universal nature, since they can be used for any working fluid (both liquids and gases).

The chief merit of these graphical symbols from a purely practical viewpoint is their simplicity. They can be rapidly drawn since they consist of lengths of straight lines, circles or arcs and arrows for showing the direction of fluid flow. They can be drawn without the need for a compass or rule, using only a plastic stencil or template.

By properly combining the basic and supplementary symbols, it is possible to depict functional graphical representations of complex apparatus.

Hydraulic graphical symbols are not restricted by any stipulated scales; in each case, the sizes of the symbols are selected to suit the size of the general diagram of the machine tool.

A system of standard graphical symbols for hydraulic apparatus was proposed a few years ago by the Joint Industry Conference of the USA. These

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TABLE 6
Hydraulic Fluid Lines and Connections

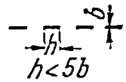
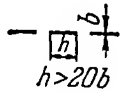
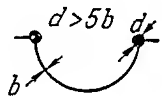
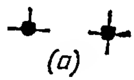
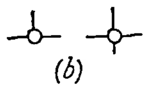
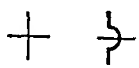
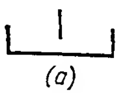
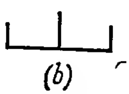
Item	Name	Symbol
1.0	Operative (working), pressure or suction lines. Piping transmitting the working fluid to the hydraulic motor or pump	
1.1	Drain, exhaust lines. Piping serving for draining, returning the working fluid back to the tank (reservoir)	
1.2	Pilot lines. Piping transmitting the working fluid for operating control units	
1.3	Flexible line. Flexible hose connected to travelling units of the machine tool	
1.4	Permanent connection of fluid lines (*a*) Separable connection of fluid lines (*b*)	 
1.5	Passing (not connected) lines	
1.6	Line to a reservoir: end of line above fluid level (*a*) end of line below fluid level (*b*)	 

TABLE 6 (continued)

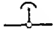
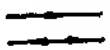
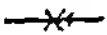
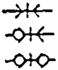
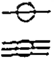
Item	Name	Symbol
1.7	Air bleeding facility	
1.8	Direction of nonreversible flow Direction of reversible flow	 (a) (b)
1.9	Plug or plugged connection for any unit or other line	
1.10	Gauge connection or pressure take-off or connection of a unit	
1.11	Fixed resistance in which the pressure drop varies with the velocity and viscosity of the fluid	
1.12	Quick connection couplings without check valves (a) with one check valve (b) with two check valves (c)	 (a) (b) (c)
1.13	Rotating joint for one or several fluid lines for one fluid line (a) for three fluid lines (b)	 (a) (b)
1.14	Telescopic connection of fluid lines	

TABLE 7

Pumps, Power Cylinders and Rotary Hydraulic Motors

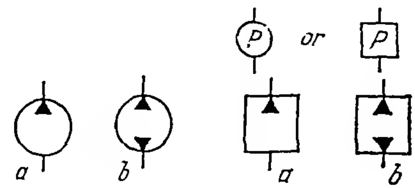
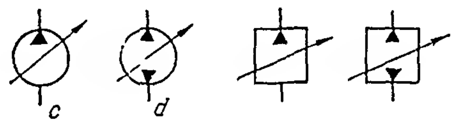
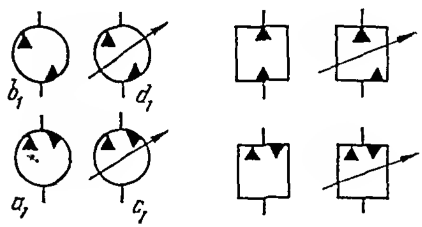
Item	Name	Symbol
2.1	Fixed-displacement pump. Heavy arrowhead shows the direction of fluid discharge from the pump: nonreversible (single-direction) pump (*a*) reversible (two-direction) pump (*b*)	
2.2	Variable-displacement pump. Heavy arrowhead shows the direction of fluid discharge from the pump: nonreversible (single-direction) pump (*c*) reversible (two-direction) pump (*d*)	
2.3	Convertible pump. Unit used either as pump or as rotary hydraulic motor in the hydraulic system. Heavy arrowheads show fluid discharge from the pump and fluid input when the pump is employed as a hydraulic motor. Fixed-displacement pump: nonreversible (single-direction) pump (a_1) reversible (two-direction) pump (b_1)	
2.4	Ditto, but variable-displacement pump: nonreversible (single-direction) pump (c_1) reversible (two-direction) pump (d_1)	

TABLE 7 (continued)

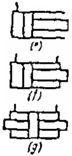
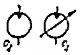
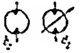
Item	Name	Symbol
25	*Single acting hydraulic cylinder* Hydraulic motor in which the pressure acts on the piston only in one direction. Piston is returned by the action of some external force or weight (*e*)	
26	Double acting hydraulic cylinder Hydraulic motor in which the pressure acts on the piston in both directions: single end rod cylinder (*f*) *double end rod cylinder (g)*	
27	Rotary hydraulic motor Unit in which energy of the fluid is converted into mechanical energy of a rotating mass. Torque on the output shaft is proportional to the capacity of the rotary motor and pressure drop between the input and output chambers (heavy arrowhead shows fluid energy input). Fixed displacement motors: nonreversible (rotation in one direction) (a) *reversible* (rotation in both directions) (b_2) Variable displacement motors: nonreversible (rotation in one direction) (c_2) reversible (rotation in both directions) (d_2)	 
28	Limited rotary hydraulic motor Unit in which the vane has a rotary oscillating motion (*h*)	

TABLE 8
Miscellaneous Units

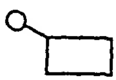
Item	Name	Symbol
3.1	Shaft of a rotary unit (electric motor, etc.). Arrow shows direction of rotation	
3.2	Component enclosure. Assembly of several units connected together by piping or internal channels in a single bank or housing, or on a single panel	
3.3	Reservoir (fluid tank) of the hydraulic system	
3.4	Manual shut-off valve	
3.5	Pressure gauge	
3.6	Electric contact with adjustable pressure (pressure switch) for operation up to a positive stop, etc. The unit has one normally open or normally closed contact controlled by the pressure in the hydraulic circuit. The dash line is the symbol for a double electric contact. The slanted arrow shows that the unit is adjustable. Other types of electric contacts are represented by symbols used in electrical engineering	 
3.7	Limit switch	

TABLE 8 (continued)

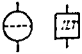
Item	Name	Symbol
3.8	Accumulator. Unit enabling a given amount of liquid to be accumulated due to deformation of elastic elements (springs, air, nitrogen, etc.), the energy being restored by discharging the liquid under pressure	
3.9	Filter. Unit designed for filtering the fluid circulating in the hydraulic system	
3.10 3.10_1 3.10_2	Electric motor fixed-speed motor *1* variable speed motor *2*	
3.11	Intensifier. Unit designed for increasing fluid pressure	
3.12	Heat exchanger. Unit used either to remove heat from the fluid—for cooling it—or for supplying heat to the fluid—for heating it. Arrows point outward for cooling and inward for heating	
3.13	Cooler. Unit used for cooling the fluid circulating in the system. Direction of arrows indicates heat removal	
3.14	Heater. Unit used for heating the fluid circulating in the system. Direction of arrows indicates *heat supply*	

TABLE 9

Directional, Reducing and Flow-Control Valves

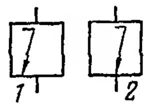
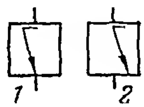
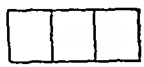
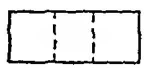
Item	Name	Symbol
4.0	The basic symbol is a square or rectangle in which lines show internal fluid flow along the channels of the unit, and an arrow shows direction of flow. Parallel shift of the lines from position to position of the unit indicates its mode of operation	
4.1 4.1_1 4.1_2	Normally closed single-channel valve: in initial position—nonoperating *1* in on position—operating *2*	
4.2 4.2_1 4.2_2	Normally open single-channel valve: in initial position—operating *1* in on position—nonoperating *2*	
5.1	Basic outline of a valve symbol. The number of squares equals the number of valve positions. Three-position valve is shown	
6.1	Symbol of a directional valve having intermediate positions is divided into squares by dash lines. Shown is the outline for the symbol of a three-position directional valve with transition through one intermediate position	

TABLE 9 (continued)

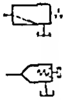
Item	Name	Symbol
7.1	Arrows show the direction of fluid circulation in the channels of the valve. In the left-hand square of the basic valve symbol, the number of lines is equal to the number of fluid lines of the hydraulic motor. In the middle square, the number of connections is equal to the number of external lines drawn to the symbol. Closed internal channels are indicated by a short line square to the fluid line. The symbol indicating reversibility is shown in the right-hand square. Combination of the right hand and middle squares characterizes the path of the fluid through the valve channels. Letter symbols: *P*—pump, *T*—tank, A_1 and A_2—hydraulic motors	
7.2	Geometrical construction of the squares of the basic symbol	
8.1	Internal connection in valve housing. Arrow shows direction of fluid discharge	
9.1	Relief valve for limiting pressure in the system to a given level	
9.2	Remotely operated relief valve (for limiting pressure)	

TABLE 9 (continued)

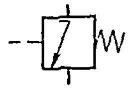
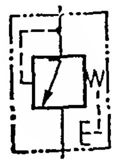
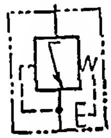
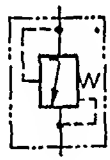
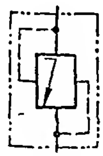
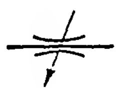
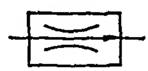
Item	Name	Symbol
9.3	Remotely operated unloading valve. Used in hydraulic systems of machines to remove pump pressure for a definite time depending upon the transmitted control signal	
10.1	Directly operated sequence valve. Used in hydraulic systems of machine tools for co-ordinating consecutive operation of two hydraulic motors, for instance, for clamping the blank followed by travel of the power piston	
11.1	Reducing valve	
11.2	Reducing valve with constant reduction. Valve enables a constant difference to be maintained between full input pressure and reduced output pressure	
11.3	Reducing valve with proportional reduction. Valve enables a constant relationship to be maintained between full input pressure and reduced output pressure	
12.1	Variable restriction (choke)	 
12.2_1	Fixed restriction	

TABLE 9 (continued)

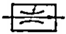
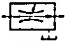
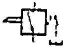
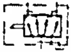
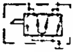
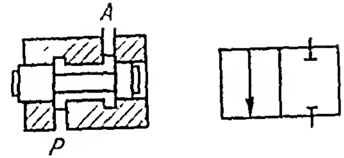
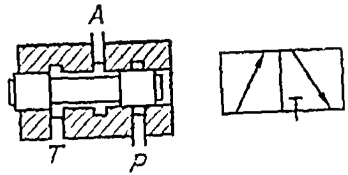
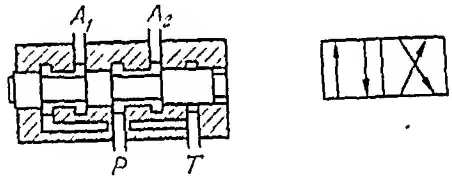
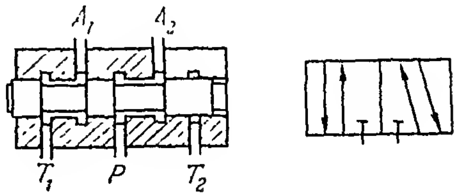
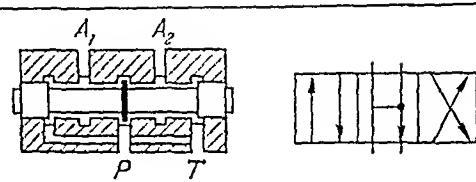
Item	Name	Symbol
12.2$_2$	Variable restriction without leakage disposal outside of valve	
12.2$_3$	Variable restriction with leakage disposal outside of valve	
12.2$_4$	Variable restriction with drain to tank for surplus fluid flowing through valve	
12.3	Normally open braking valve for hydraulic motors. The valve is mounted in the output line and is actuated by a cam which overcomes the pressure of a spring	
12.5	Tracing device with a single-edge directional valve	
12.4	Adjustable four-way open-centre directional valve	
12.6	Tracing device with a four-edge directional valve	
12.7	Adjustable four-position closed-centre directional valve	

TABLE 10
Standard Directional Valves

Item	Type	Valve	Spool position	Symbol
01	—	Two-position two-way shut off	I. P and A connected II. P and A blocked off	
02	2БГ73-2	Two-position three-way	I. T blocked off, P and A connected II. T and A connected, P blocked off	
03	Г72-1 5Г73	Two-position four-way	I. Connected: P to A_2 and A_1 to T II. Connected: P to A_1 and A_2 to T	
04	Г72-2 7Г73	Two-position five-way	I. Connected: P to A_2 and A_1 to T_1 II. Connected: P to A_1 and A_2 to T_2	
05	1Г73-1 1Г73-4	Three-position four-way	0. Connected: P to A_1 and A_2 to T I. Connected: P to A_2 and A_1 to T II. Connected: P to A_1 and A_2 to T	

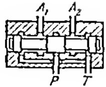
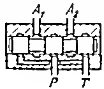
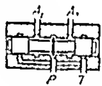
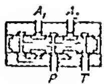
06	2173-1 2173-4	Three position four way	0 Connected T to A_1 to A_2, P blocked off I Connected P to A_1 and A_2 to T II Connected P to A_2 and A_1 to T		
07	4173-1 4173-4	Three position four way	0 P and T blocked off I Connected P to A_1 and A_2 to T II Connected P to A_2 and A_1 to T		
08	3174-1 3174-[illegible]	Three position four way	0 T blocked off, connected A_1 to P and A_2 to P I Connected P to A_1 and A_2 to T II Connected P to A_2 and A_1 to T		
09	[illegible]173-1	Three position four way	0 Connected P to T, A_1 and A_2 blocked off I Connected P to A_1 and A_2 to T II Connected P to A_2 and A_1 to T		

TABLE 11

Examples of Symbols Representing Combination Controls and Certain Circuits of Units

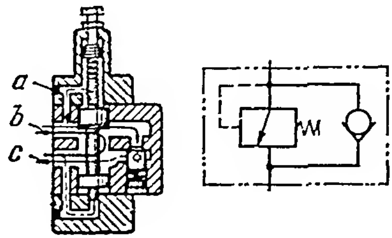
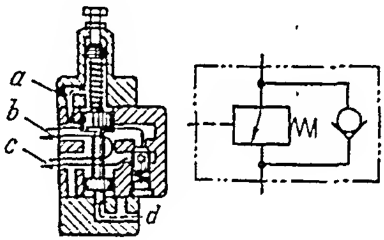
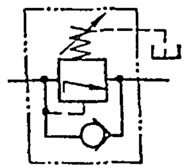
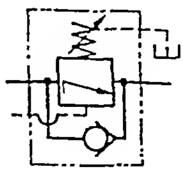
Name	Symbol
Internally piloted counterbalance valve with integral check valve, model Г66-2: *a*—internal seepage drain; *b*—outlet to system and outlet in reversing; *c*—inlet to valve and outlet in reversing	
Remotely piloted counterbalance valve with integral check valve: *a*—internal seepage drain; *b*—outlet to system and inlet to valve in reversing; *c*—inlet to valve and outlet in reversing; *d*—remote pilot connection	
Internally piloted, normally closed sequence valve with adjustable spring and integral check valve	
Remotely piloted, normally closed sequence valve with adjustable spring and integral check valve	

TABLE 11 (continued)

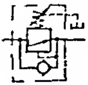
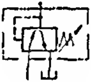
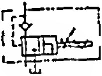
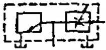
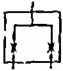
Name	Symbol
Adjustable normally open reducing valve with integral check valve, for reverse travel of the hydraulic motor	
Adjustable normally closed reducing valve that unloads to tank when reduced pressure is exceeded	
Automatic valve for switching pump on and off during operation with an accumulator	
Flow control valve with maximum pressure and external drain	
Fluid flow divider valve for synchronizing travel of hydraulic motors	

TABLE 11 (continued)

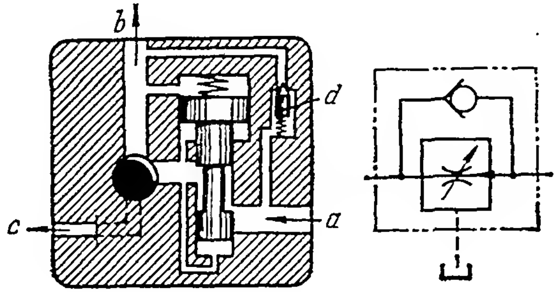
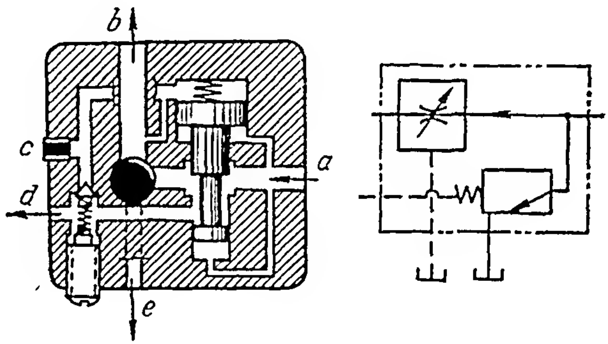
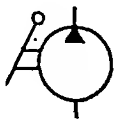
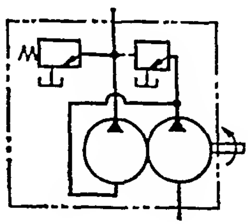
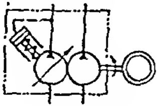
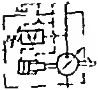
Name	Symbol
Pressure-compensated flow-control valve for one direction of flow with check valve for reversing: *a*—inlet; *b*—outlet to system; *c*—external seepage drain; *d*—check valve	
Pressure-compensated flow-control valve for one direction of flow with relief valve for limiting reduced pressure: *a*—pressure inlet; *b*—reduced pressure outlet; *c*—outlet to unloading valve; *d*—outlet to tank; *e*—external seepage drain	
Manually controlled variable-displacement pump	
Fixed-displacement pumps connected in series with maximum pressure at output and remote control of the pressure of the first-stage pump	

TABLE II (continued)

Name	Symbol
One fixed- and one variable-displacement pump connected in parallel. Automatic control of variable-displacement pump in accordance with developed pressure, thereby leading to constant consumption of maximum power	
Variable-displacement pump with servomotor controls. Control valve is mechanically actuated by a cam. Fixed-displacement, constant pressure pump is used in the servomechanism	
Variable-displacement pump with servomotor controls actuated by a cam. Control mechanism has a differential piston	
Variable-displacement pump controlled from the pressure which is preset by adjusting the spring of the control piston	

and the required power rating of the drive motor. The data on variable-displacement pumps include the delivery range.

Machine tools with a power consumption $N \leqslant 3$ kW, usually have double or single vane pumps developing a pressure $p \leqslant 55$ kgf per sq cm. In the USSR, these are single vane pump, model Г12-1, with a delivery in the range $Q = 5$ to 200 litres per min, and double vane pump, model 5Г12-1, with various combinations of deliveries.

Variable-displacement rotary piston pumps, developing a pressure $p = 100$ to 200 kgf per sq cm, are to be preferred for machine tools with a power consumption $N \geqslant 5$ kW (broaching machines, planers, slotters, etc.).

The power consumption of machine tools operating at medium and high speeds, with a small pulling force (grinding, lapping and honing machines) is usually limited to 4 kW and the fluid pressure does not exceed 20 kgf per sq cm. Gear pumps, models Г11-1 and Г11-2, with a delivery range from 12 to 140 litres per min, find application in these machines.

The diameter of the power cylinder of the machine tool is selected in accordance with the force that the piston must exert, or with its required speed. In some cases, other factors, such as requirements as to smooth travel, overall size, stability of travel, etc., influence the choice of the cylinder diameter. If the pressure in the cylinder is increased, its diameter is reduced as are the required pump output, pipe size and the size of all the other hydraulic apparatus. As a result, a more compact system is obtained, that is easier to assemble and install. On the other hand, a higher pressure requires the use of more expensive pumps, higher leakproof qualities of the piping and more reliable packing and seals.

The pressure in the power cylinder does not, as a rule, exceed 45 kgf per sq cm.

A ratio of the cylinder length l to its diameter D over 20 is not advisable. It has been observed that at $l > 20\,D$, there is a danger of self-induced vibration due to the increase in the elasticity of the hydraulic system. Hence, instead of long reciprocating strokes of a power piston, it will prove more expedient to use a rotary hydraulic motor, capable of developing the required torque, in conjunction with a rack-and-pinion or power screw drive. The additional train of gears from the rotary motor shaft to the table rack increases the cost of the machine, reduces its efficiency and increases the inertia in comparison with a system based on a power cylinder.

Standard control apparatus, standard pipe fittings and connections, and standard seals and packing should be used throughout in designing hydraulic circuits. The construction of these elements has been developed by ENIMS and has found extensive application in Soviet machine tool production.

The hydraulic circuit diagram of a machine tool should be as illustrative as possible and easy to read. A diagram with these features can be made if

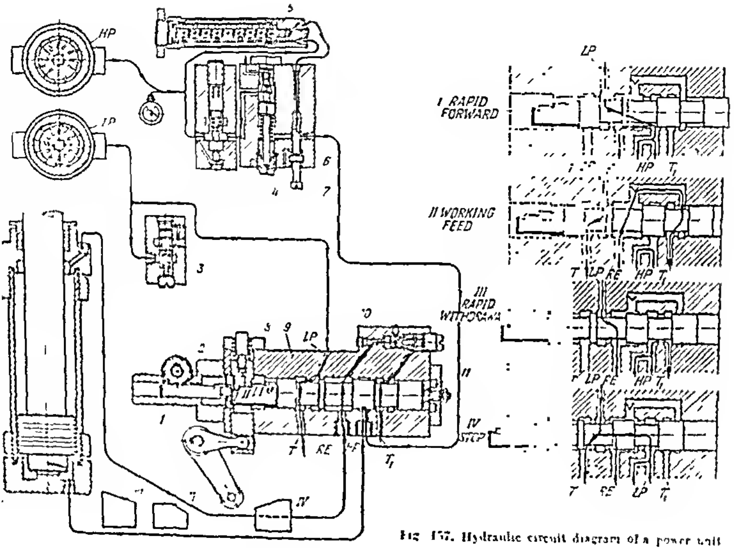


Fig. 157. Hydraulic circuit diagram of a power unit

and the required power rating of the drive motor. The data on variable-displacement pumps include the delivery range.

Machine tools with a power consumption $N \leqslant 3$ kW, usually have double or single vane pumps developing a pressure $p \leqslant 55$ kgf per sq cm. In the USSR, these are single vane pump, model Г12-1, with a delivery in the range $Q = 5$ to 200 litres per min, and double vane pump, model 5Г12-1, with various combinations of deliveries.

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The power consumption of machine tools operating at medium and high speeds, with a small pulling force (grinding, lapping and honing machines) is usually limited to 4 kW and the fluid pressure does not exceed 20 kgf per sq cm. Gear pumps, models Г11-1 and Г11-2, with a delivery range from 12 to 140 litres per min, find application in these machines.

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The pressure in the power cylinder does not, as a rule, exceed 45 kgf per sq cm.

A ratio of the cylinder length l to its diameter D over 20 is not advisable. It has been observed that at $l > 20\,D$, there is a danger of self-induced vibration due to the increase in the elasticity of the hydraulic system. Hence, instead of long reciprocating strokes of a power piston, it will prove more expedient to use a rotary hydraulic motor, capable of developing the required torque, in conjunction with a rack-and-pinion or power screw drive. The additional train of gears from the rotary motor shaft to the table rack increases the cost of the machine, reduces its efficiency and increases the inertia in comparison with a system based on a power cylinder.

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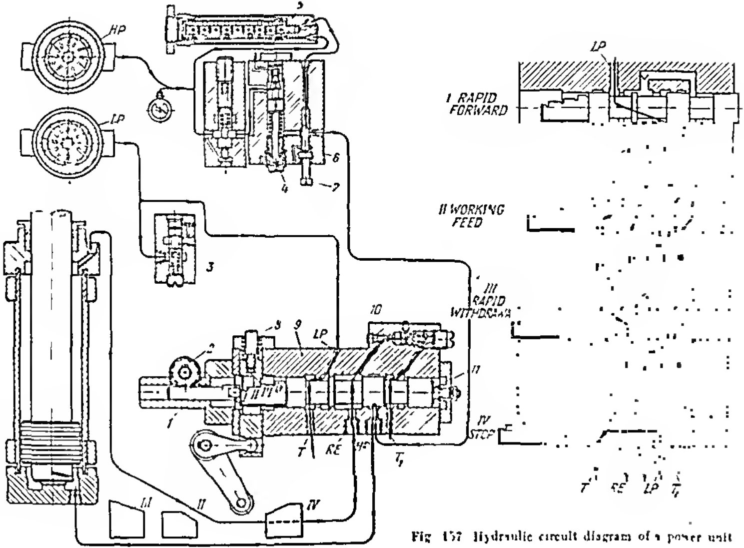

Fig 157 Hydraulic circuit diagram of a power unit

the conventional representations or graphical symbols are properly arranged with the minimum number of intersecting lines.

In the Soviet machine tool industry, the Stankoprinadlezhnosti Plant (Hydroprivod and SDO-7) is engaged in the centralized manufacture of standard hydraulic control panels (types У4251, У424, 3У4222, У4244, У4245, У4246, 3У4252 and У4247) for various cycles of drilling, boring and milling operations with remote and automatic control systems. The plant also produces hydraulic control panels for grinding machines (models ГШ, ГШП, ГШУ and Г31-1), designed for various operating cycles.

An example of a control panel with automatic command signal change-over is shown schematically in Fig. 157. The panel is designed for operation with two fixed-displacement pumps, one for low and the other for high pressure. The low-pressure pump *LP* is used for rapid approach and withdrawal movements of the power piston; the high-pressure pump *HP* is for working travel. The cycle provided by the panel constitutes: rapid approach—working feed—rapid withdrawal—stop.

Main directional control valve *9*, flow-control valve *6* of power piston travel and relief valve *3* of the low-pressure pump are mounted on a plate.

Oil delivered by the pumps is distributed by the main valve. The positions of the valve spool for the various command signals are shown separately.

The following are used for rapid forward travel: oil exhausted from the rear end of the cylinder and oil delivered by the low-pressure pump *LP*. The output of pump *HP* is unloaded at low pressure to the tank through valve *4*.

During working feed, oil from pump *LP* drains back to the tank at zero pressure along line *LP-T*. Oil is delivered from the high-pressure pump to the front end of the cylinder through flow control valve *6*.

The rear end of the cylinder is connected to backpressure valve *10* ($p = 2$ to 5 kgf per sq cm). This valve serves to equalize the friction forces to some degree.

Upon rapid withdrawal, oil from the low-pressure pump is delivered to the rear end of the cylinder through line *LP-RE*. At this, the front end is connected to the tank. In the STOP position, both pumps and both ends of the cylinder are connected to the tank.

The spool of the main valve is shifted to the RAPID FORWARD position manually by means of pinion *2* and rack *1*. This motion compresses spring *11* in the valve, thereby providing for return of the spool.

As the control lever runs up on cams *II*, *III* and *IV*, detent *8* is lifted, enabling the valve spool to shift by spring action to the positions: WORKING FEED, WITHDRAWAL and STOP. Since the spool must pass through the WORKING FEED and WITHDRAWAL positions when it is again shifted to the RAPID FORWARD position, the lever (linked to

pinion 2) must be turned rapidly to avoid motion of the power piston in the intermediate positions

Since the opening in throttle 7 is very small (fraction of a square millimetre), the oil is passed through filter 5 to avoid clogging the throttle

The relief valve of the high pressure pump is set to a pressure from 5 to 10 kgf per sq cm higher than that required in the machining process This adjustment is made with the power piston up against a positive stop The relief valve of the low pressure pump is adjusted to the pressure required for rapid traverse movements The pressure gauge shows the pressure in the system at all times except during the return stroke

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